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OPTICAL RADAR ANGLE TRACKING MOUNT

George J. Thompson, et al

Owens-Illinois, Incorporated

Prepared for:

Rome Air Development Center

July 1973

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Technical Report
July 1973

OPTICAL RADAR ANGLE TRACKING MOUNT

Owens-Illinois

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Rome Air Development Center
Air Force Systems Command
Griffiss Air Force Base, New York

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George J. Thompson
Spiro Pappas

Owens-Illinois

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This report was partially funded
under ARPA Order 1279

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FOREWORD

This interim technical report, covering the period April 1972 to May 1973, was prepared by Owens-Illinois, Incorporated, Fecker Systems Division, Pittsburgh, Pennsylvania, under contract F30602-72-C-0192, Job Order Number 65270121, with Rome Air Development Center, Griffiss Air Force Base, New York. The investigation is also partially sponsored by the Defense Advanced Reseach Project Agency under ARPA Order 1279.

This report was numbered by Owens Illinois F(4)-864-047-022-2251. Investigator for the technology contained herein was Spiro Pappas.

This report has been reviewed by the RADC Information Office (OI) and is releasable to the National Technical Information Service.

This technical report has been reviewed and is approved.

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A Technical Report designated F(4)-864-022-2251 (RADC-TR No. 1279) consisting of an Environmental Analysis in two parts: Random Solar Heat Pointing Error; and the Hydrostatic Bearings Design in three parts: Azimuth Axis Thrust Bearing, Azimuth Axis Radial Bearing, and Evaluation Axis Radial Bearing. These subjects are critical design criteria for the Optical Radar Angle Tracking Mount.

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ABSTRACT

A Technical Report designated F(4)-864-047-022-2251 (RADC-TR No. 1279) consisting of an Environmental Analysis in two parts, Random Solar Heat Pointing Error and Oil Bearing Heat Pointing Error and the Hydrostatic Bearings Design in three parts, Azimuth Axis Thrust Bearing, Azimuth Axis Radial Bearing and Elevation Axis Radial Bearing. These subjects are critical design criteria for the Optical Radar Angle Tracking Mount.

TABLE OF CONTENTS

SECTION I

ENVIRONMENTAL ANALYSIS

1. INTRODUCTION

The following environmental conditions, in which the Optical Radar Angle Tracking Mount must operate, are restated below (from PR No. A-1-1095, paragraph 3.6):

Temperature -

Operating:	-20° F to + 100° F
Non-Operating:	-35° F to + 140° F
Stability:	10° F change in a 60 minute period

Humidity -

Operating or non-operating: 10 to 100% Rh with condensation due to temperature changes.

a. Authorization

The following theoretical environmental analysis is in lieu of actual environmental testing and is included in this Technical Report by authorization of PR No. A-1-1095, Amendment #1, 21 December 1971, attachment #2.

b. Scope

Regarding overall temperature range or temperature changes in a relatively short period of time such as 2 or 4 hours maximum, only hot spots or non-uniformly heated (or cooled) areas are considered as contributions to the pointing error of the mount. Uniform heating (or cooling) regardless of rate of temperature change causes no calcuable distortions or alignment alterations since material selections are compatible throughout the instrument.

Non-uniform heating does, however, exist and manifests itself in two distinct modes: first, externally from random solar heating, and second, internally from oil bearing heat. Both are discussed herein in paragraphs I.2 and I.3, respectively.

Regarding relative humidity or condensation resulting from coincident temperature and humidity conditions, all exposed surfaces of the entire mount are painted, plated or protected in some way such that condensation will cause no detrimental effect, with one exception, that being the elevation mirror surface. Any contamination that may fall upon the mirror surface while it is wet will be left behind when the water dries, leaving a film which can reduce the reflectivity of the mirror. Periodic surface cleaning can correct this

problem, or some steps can be taken to prevent the problem. First, always store the elevation mirror, when not in use, at either 0° or 180° of Elevation Angle. This will put the plane of the mirror vertical. Second, during climatic periods of high humidity and low temperature, warm the ambient air inside the dome and, if possible, refrain from use in order to keep the dome closed.

2. RANDOM SOLAR HEAT POINTING ERROR

When the coelostat is operating on a sunny day with the dome doors open, the radiant energy of the sun will not uniformly bathe the entire instrument. The solar radiation will fall on different areas and even move across these areas when the mount is moving. Most of the areas are either massive and require much time to absorb significant quantities of heat or would not create an error contribution even when their dimensions become altered by a temperature increase. One area is critical however, that being the vertical support column for the elevation axis. An increase in column length results to produce a non-systematic pointing error by altering the orthogonality (90° angle) between the elevation and azimuth axes.

Referring to Appendix A, the following values can be concluded:

a) that the temperature rise in the support column can be 8° F.
b) that the elongation of the column due to this 8° F change can be .00192 inches.
c) that the resultant orthogonal angle can vary by 27.4×10^{6} rad (5.66 arc seconds).
d) that these changes can take place in 42 minutes.
e) finally, that the additional orthogonality error can not be tolerated and must be eliminated or reduced to an insignificant value.

a. Solar Radiation Shield

Insulation can be applied to shield the outboard elevation support column from the radiant energy of the sun.

The shield consists of a protective enclosure, which shields the four sides of the column which may be exposed to solar radiation. The exposed outside surface of the enclosure is lined with a sheet of aluminized mylar to reflect solar radiation. The mylar is bonded to a wall of rigid polyethylene foam of which the enclosure is fabricated. The polyethylene also acts to retard any heat that is not reflected by the aluminized surface. The aluminized surface reflects approximately 90 - 95% of the incident solar radiation.

In mounting the enclosure into position, a 2-inch air gap is provided between the enclosure wall and the support column to promote convection. This is a source of further heat removal. It is expected that the combination of reflection, retardation, and convection will be 99.9% effective, thus eliminating the solar radiation effects on the orthogonality portion of pointing error.

3. OIL BEARING HEAT POINTING ERROR (See also Appendix B)

Heat in a hydrostatic bearing results primarily from hydraulic pressure energy which is converted directly into heat energy when a pressure drop occurs thru the compensating elements and the bearing pads. No work is performed by this pressure drop so that the energy conversion into heat is virtually complete. This heat will be transferred to the rotor and stator of the bearing, and then, further, transferred to the structure local to the bearing. From this heat, dimensional changes will occur.

At the azimuth oil bearing, dimensional changes would be symmetrical both in radial and axial directions and thus cause no angular variations and therefore no pointing error contributions.

At the elevation oil bearing, however, since the axis of the bearing (Elevation axis) is horizontal, a radial dimensional increase will move the bearing axis a distance equal to 1/2 the total diametral change, which results to produce a non-systematic pointing error by altering the orthogonality (90° angle) between the elevation and azimuth axes.

Referring to Appendix B, the following values can be concluded:

a) Pointing Error $\Delta\theta = 2.5 \times 10^{-6}$ rad/$^{\circ}$F.
(or 0.56 arc sec/$^{\circ}$F).

b) Conservative maximum temperature variation T3-Ta = 2.4°F.

c) A realistic value for T3-Ta is 1.6°F.

d) The orthogonality can vary by $\pm 2.0 \times 10^{-6}$ rad ($\pm$.41 arc seconds).

e) Finally, that the additional orthogonality error can be tolerated provided that the total system pointing error does not exceed the specification.

a. Oil Cooler

If the total system pointing error approaches the specification value, some help can be offered through the use of a heat exchanger in the hydraulic supply system. This cooling unit could be installed in the oil return line at the location of the hydraulic power unit. There are, however, several other natural conditions for heat exchange which preclude the use of the cooler.

SECTION II

HYDROSTATIC BEARINGS DESIGN

1. INTRODUCTION

Three hydrostatic oil bearings are used in the design of the optical radar angle tracking mount. The bearings provide smooth, precise, and friction free operation throughout the operational range of the instrument. The bearings include the azimuth axis thrust bearing, the azimuth axis radial bearing, and the elevation axis radial bearing. The azimuth axis bearings are combined into one assembly to provide total restraint capability for the azimuth axis. Oil is supplied to all three bearings at 230 psi.

Operational temperature limits of the coelostat are between -20° F and $+ 100^{\circ}$ F. Bearing oil viscosity will change accordingly. If the viscosity is too high, it will have difficulty in being pumped. If the viscosity is too low the oil will be too thin and excessive flow will result. In the present application, the recommended viscosity limits for bearing operation are 4×10^{-6} REYN to 28×10^{-6} REYN (32-200 Centistokes). One single fluid cannot provide these narrow viscosity limits in the temperature range -20° F to $+ 100^{\circ}$ F. Two fluids are therefore recommended with change-over periods occurring in the Fall and Spring of each year. During cold weather operation (Nov. 15 - April 30) the operational fluid will be MIL-H-5606B (Exxon Univis J41). During warm weather operation (May 1 - Nov. 14) the operational fluid will be automatic transmission fluid (Exxon Glide or equivalent).

2. DESIGN REVIEW

During the design phase of this hardware contract, a consultant was retained to review, recommend, and contribute if necessary to these hydrostatic bearing designs. A copy of their report follows as Exhibit A. The three potential problem areas mentioned by Mr. Rippel in the Franklin Institute report were treated as follows:

EXHIBIT A

THE FRANKLIN INSTITUTE
RESEARCH LABORATORIES

THE BENJAMIN FRANKLIN PARKWAY • PHILADELPHIA, PENNSYLVANIA 19103 • TELEPHONE (215) 448-1000

MECHANICAL AND NUCLEAR ENGINEERING DEPARTMENT

August 16, 1972

Owens-Illinois Inc.
Fecker Systems Div.
4709 Baum Blvd.
Pittsburgh, Pa. 15213

Attention: Mr. Spiro Pappas

Subject: FIRL evaluation of Owens-Illinois design approach of oil hydrostatic bearings for a 2-axis optical radar tracking mount.

References: 1. Owens-Illinois Purchase Order No. C31340-9-864 dated 8/10/72

2. Engineering discussions held at Owens-Illinois Fecker Systems Div. Pittsburgh facility on August 14 & 15, 1972

Gentlemen:

The purpose of this letter is to formalize the writer's evaluation of the Owens-Illinois oil-lubricated hydrostatic bearing design approach for application to a 2-Axis Optical Radar Tracking Mount. This evaluation was performed for Owens-Illinois (Ref. 1) during recently held discussions with cognizant O-I personnel at your facility (Ref. 2)

Oil-lubricated hydrostatic bearings are to be applied to the azimuth axis and to the elevation axis (inboard end only) of the subject mount. The azimuth bearing consists of three bearing surfaces, namely, upper and lower thrust surfaces and a radial bearing surface. All such bearing surfaces are of the multi-pad type. The number of pads used and their locations are such as to provide the necessary thrust, radial and moment load-carrying capability. The elevation axis bearing is also a multi-pad journal bearing as required for support of the radial load.

The bearing analysis methods used are adequate for the bearing concepts employed. Hence, predictions of oil-film load-carrying capabilities, stiffnesses, flows, pressures, etc., are valid.

The subject application uses the capillary-compensated, constant pressure source type of lubricant supply distribution circuit. This type of circuit minimizes the changes in bearing performance due to the broad range of operating temperature.

The seemingly novel method of providing pressurized flow to the elevation bearing (via a manifold incorporated within the azimuth radial bearing) is a well proven technique. In effect, a "hydraulic slip ring" is realized that is similar in principle and in form to that used in various engine crankshafts to supply pressurized oil from the rotating crankshaft to lubricate the oscillating connecting rod wrist pin bearings.

The writer is in agreement with the design modification to supply both bearings with the same value of supply pressure (230 psig). Thus, the major pressure-drop in the elevation bearing will be across its capillary tube restrictors (230-35 = 195 psig). This will result in about a 60% increase in elevation bearing oil-film stiffness compared to that realized with only a 70 psig supply pressure.

The selection of oil-film clearances in the range of 0.002 to 0.003 inch is within the right range from the competing considerations of flows required, oil-film stiffnesses, and required bearing fabrication and installation tolerances.

In our evaluation of the design approach (in its current state of completeness) we would foresee a number of potential problem areas, namely:

- Adequate and purposeful control of supply oil temperature to minimize deleterious thermal gradients within the mount.
- The incorporation of as much lubricant drainage as is physically possible in order to promote the free flow of return oil (by gravity) back to the reservoir(s) without the build-up of excessive back pressures in the main discharge oil collecting cavities.
- Matching of structural compliance characteristics of mating bearing member support structures to insure against loss of bearing oil-film performance due to compliance-induced misalignments of bearing surfaces.

All of these potential problem areas are recognized by O-I and are, therefore, being given careful consideration during the bearing design phase now underway.

In summary then, on the basis of our limited evaluation it is our opinion that the O-I hydrostatic bearing design effort is proceeding in the right direction and is backed-up with sufficiently realistic analysis methods and capabilities that are required to generate sufficient confidence that the final bearing design to be applied will provide adequate performance before said design is committed to hardware.

The FIRL is pleased to have been of service to Owens-Illinois. We shall be pleased to provide additional services if and as subsequently required during the course of completion of the bearing design effort by Owens-Illinois.

Sincerely,

H. C. Rippel

H. C. Rippel
Principal Engineer

kas

cc: Mr. G. J. Thompson
Project Engineer
Owens-Illinois

Mr. John Daker
Purchasing Agent
Owens-Illinois

a. Oil cooling received the fullest consideration in the final bearing designs but the decision for an oil cooler is still held in abeyance not because of any bearing performance characteristic, but because of the possibility of the oil bearing heat pointing error contribution which is discussed in paragraph 1.3 of this report.

b. Maximum possible drainage ducts, within space allowances, were provided, especially from the elevation bearings. In addition, a vacuum scavenge system has been included to assist in the return oil flow and prevent oil seepage through the gaps between the rotor and stator of each bearing. This vacuum scavenge system will cause a pressure differential across the oil bearing gaps of about 1 psi.

c. Matching structural compliances were improved in the following manner to insure against loss of bearing oil-film performance due to compliance induced mis-alignments of bearing surfaces:

- Stiffened structure
- Stiffened rings
- Elevation housing wall thickness was increased
- O.D. of azimuth bearing supports were increased

3. AZIMUTH AXIS THRUST BEARING (See also Appendix C)

The azimuth axis thrust bearing provides thrust support for the total moving mass of coelostat and also provides restraining moment for the axis about a plane perpendicular to the azimuth axis ("Rocking Moment"). The bearing consists of twelve equally spaced pairs of opposed locating pads (total of 24 pads) arranged in a circle which has a mean radius of 26.88 inches. An oil supply pressure of 230 psi is applied through twenty-four capillary compensating tubes for regulation of oil flow to the individual pads. The nominal preload clearance is 0.0025 inches. The applied load produces an eccentricity of approximately 0.000675 inches. Rocking moment stiffness exceeds 20×10^9 in-lbs/rad which is more than adequate.

4. AZIMUTH AXIS RADIAL BEARING (See also Appendix D)

The azimuth axis radial bearing is a journal bearing which restrains any lateral movement by the azimuth axis. The bearing is 58.00 inches in diameter and consists of twelve equally spaced pads pre-loaded against the journal to provide the required stiffness. The supply pressure is 230 psi (as in the other two bearings) and is equally distributed to the pads by twelve capillary flow

restrictors, one for each pad. The nominal radial clearance (and film thickness) is 0.0025 inches. Axis eccentricity can only occur with the introduction of a side load. Stiffness of the bearing, however, is quite high and any resulting eccentricity would be insignificant.

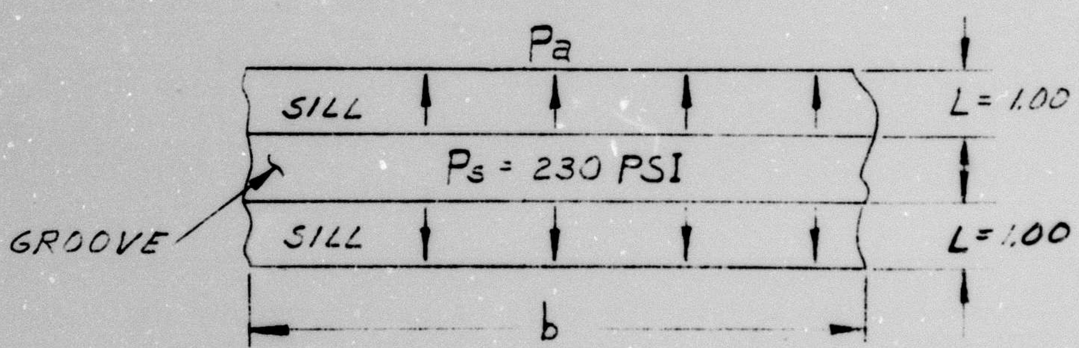
5. ELEVATION AXIS RADIAL BEARING (See also Appendix E)

The elevation bearing is a journal bearing which supports the inboard side of the elevation axis. The bearing has eight equally spaced hydrostatic pads which are an integral part of the outer support ring. The journal diameter is 55.00 inches. Bearing oil supply pressure is 230 psi and is applied across eight capillary flow regulators to properly distribute the oil to the eight individual bearing pads. The no-load radial clearance between journal and support ring is 0.003 inch. Application of the bearing load introduces an eccentricity of 0.0009 inches.

6. BEARING OIL TRANSFER SLOT

Providing oil under pressure to the inner axis bearing from an oil supply system externally located is accomplished by the use of an oil transfer slot. The transfer slot is considered a hydraulic slipring, but with some leakage. The oil passes from a stationary ring to a rotating ring having a circumferential groove. The groove serves as a common supply for the eight capillary inlets of the elevation bearing. The leakage is controlled by fixing the sill length and cap of the leakage path. The gap or radial clearance is 0.0025 inches and the length of the sill is 1.00 inch. Flow across the slot (leakage) is hence laminar and can be determined with the following flow equation:

$$Q = \frac{\Delta p\, h^3 b}{12 \mu L}$$



Where:

Q = Flow (in^3/sec)
b = Sill width (ring circumference) inch
L = Sill length (inch)
h = Film thickness (gap) inch
μ = Viscosity (REYN)
ΔP = Ps - Pa (PSI)
Ps = Supply pressure (PSI)
Pa - Ambient pressure (PSI)

$$\Delta p = 230 \text{ PSI}$$
$$L = 1.00 \text{ inch}$$
$$b = \pi \times (58.00) = 182.2 \text{ inch}$$
$$h = 0.0025 \text{ inch}$$
$$\mu = 4 \times 10^{-6} \text{ REYN}$$

$$Q = \frac{230 \times 0.0025^3 \times 2 \times 182.2}{12 \times 4 \times 10^{-6} \times 1.0}$$

$$Q = 27.2 \text{ in}^3/\text{sec}$$

Flow Horsepower

$$Hp = \frac{27.2 \times 230}{6600}$$

$$Hp = 0.95 \text{ hp}$$

7. CAPILLARY COMPENSATION

Compensating elements for hydrostatic bearings - flow restrictors between the oil supply pump and the bearing pads - have a definite effect on bearing performance. Capillary compensation is used in this application because it offers several advantages. It is desirable to maintain the pressure ratio, pad stiffness and bearing eccentricity invariant with variations in bearing fluid viscosity (which will occur with variations in temperature). Only capillary compensation will provide this advantage. This phenomenon is particularly important in the elevation axis bearing whose axis is horizontal. Changes in bearing eccentricity have a marked effect on orthogonality to the azimuth axis, and hence the accuracy or pointing error of the instrument.

8. SUMMARY

Included in Table I is a summary of parameters for the three hydrostatic oil bearings and the oil transfer slot.

TABLE I

SUMMARY OF PARAMETERS

PARAMETER / BEARING	AZIMUTH THRUST BEARING	AZIMUTH RADIAL BEARING	ELEVATION RADIAL BEARING	OIL TRANSFER SLOT
Bearing Load (lb)	39,000	0	5000	
Number of Pads	24	12	8	
Pad Area - A_p (in.^{2})	48.7	27.0	60.8	
Pad Area Coefficient - a_f	0.743	0.64	0.689	
Max Bearing Flow - Q (in.^{3}/sec)	29.8	6.12	3.67	27.2
Flow Coefficient - q_t	3.90	2.95	3.40	
Supply Pressure - Ps (PSI)	230	230	230	230
Preload Press Ratio - B'	0.50	0.30	0.15	
Pad Preload - W' (lb)	4160	1192	1463	
Preload Clearance - C (in.)	0.0025	0.0025	0.003	0.0025
Eccentricity - e (in.)	0.000675	0	0.0009	
Eccentricity Ratio - ε	0.27	0	0.30	
Bearing Stiffness - (lb/in.)	4.51×10^6*	6×10^6	5.5×10^6	
Capillary Tube Diam - d_c (in.)	0.054	0.035	0.035	
Capillary Tube Length - l_c (in.)	4.63	2.91	4.00	
Flow Hp - @ Max Flow	1.04	0.214	0.128	0.95

*Per Pad Pair - Moment Stiffness is 24×10^9 in-lb/rad

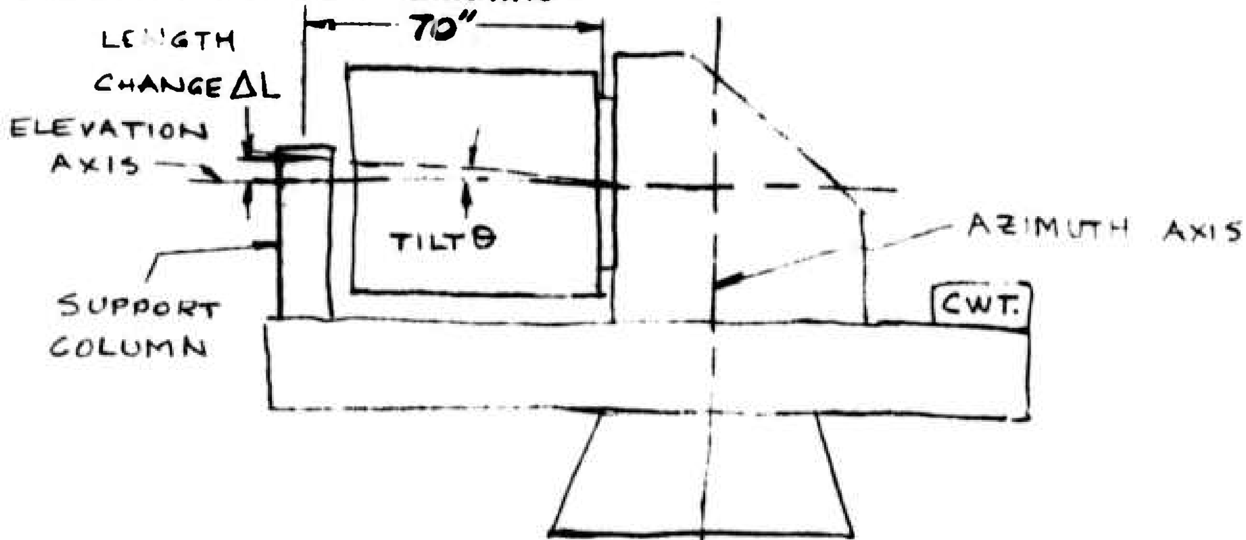
APPENDIX A

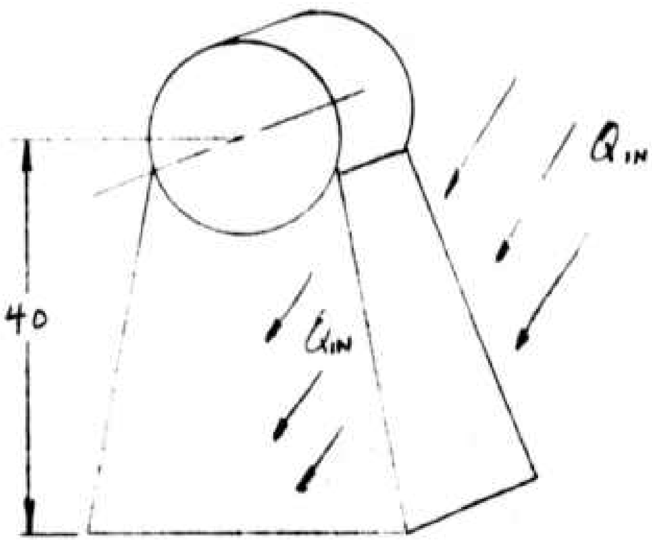
RANDOM SOLAR HEAT POINTING ERROR

RANDOM SOLAR HEAT POINTING ERROR

THE COELESTAT IS SUSPENDED IN AN AZIMUTH OVER ELEVATION MOUNT. DURING OPERATION, SOLAR RADIATION MAY CAUSE NON SYMMETRIC DISTORSIONS OF THE MOUNT AND SUBSEQUENT POINTING ERROR. THE PURPOSE OF THIS ANALYSIS IS TO DETERMINE THE POSSIBLE POINTING ANGLE CHANGE WHICH COULD RESULT FROM SOLAR RADIATION.

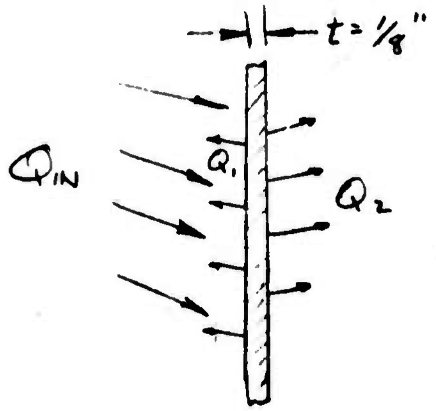
THE AFFECT OF SOLAR RADIATION IS TO NON-SYMMETRICALLY HEAT THE SUN SIDE OF THE STRUCTURE WHILE THE OPPOSITE SIDE REMAINS COOL AND UNAFFECTED. OF THE VARIOUS NON-SYMMETRICAL MODES, THE ONE MUST LIKELY AND SIGNIFICANT IS THE TILTING OF THE ELEVATION AXIS (ORTHOGONALITY ERROR) AS A RESULT OF SOLAR RADIATION IMPINGING ON THE OUTBOARD ELEVATION BEARING SUPPORT COLUMN.





ELEVATION SUPPORT COLUMN

FOR PURPOSES OF ANALYSIS, THE SUPPORT COLUMN IS ASSUMED TO BE A RECTANGULAR BOX LIKE STRUCTURE HAVING THIN WALL SIDES 1/8" THICK.



TYPICAL WALL
RECEIVING AND LOSING HEAT ENERGY

ASSUME THAT TWO WALLS RECEIVE SOLAR RADIATION AND LOSE IT AS FOLLOWS

Q_{IN} - SOLAR RADIATION ABSORBED BY WALLS

Q_1 - CONVECTION LOSSES FROM OUTER SURFACES (BTU/HR)

Q_2 - CONVECTION LOSSES FROM INNER SURFACES (BTU/HR)

Q_3 - RADIATION LOSSES FROM OUTER SURFACES (BTU/HR)

Q_4 - RADIATION LOSSES FROM INNER SURFACES (BTU/HR)

NEGLECTING RADIATION LOSSES Q_3 AND Q_4 AND ASSUMING UNIFORM TEMP ACROSS THE WALL THICKNESS

$$Q_{IN} = Q_1 + Q_2$$

$$Q_1 = h_1 A (T_f - T_o)$$

$$Q_2 = h_2 A (T_f - T_o)$$

h_1 AND h_2 ARE THE FILM COEFFICIENTS OF THE OUTER AND INNER WALL (BTU/FT^2-HR-°F)

T_f - FINAL EQUILIBRIUM TEMPERATURE OF THE HEATED SURFACE (°F)

T_o - SURFACE TEMPERATURE PRIOR TO EXPOSURE TO SUNLIGHT (°F) (ALSO ASSUMED AMBIENT)

A - CHARACTERISTIC AREA (FT^2)

$$Q_{IN} = \bar{\alpha}\, \bar{Q}\, A$$

$\bar{\alpha}$ = SURFACE ABSORPTIVITY

$\bar{Q}$ = SOLAR RADIATION BTU/HR-FT^2

$\bar{\alpha} = 0.5$

$\bar{Q} = 160$ BTU/HR-FT^2 CONSERVATIVE VALUE - FROM KENT - "MECH. ENG. HANDBOOK"

IF, HOWEVER, (1) ONLY THE PROJECTED AREA OF 2 SIDES IS CONSIDERED RATHER THAN THE TOTAL AREA (2) THE SUPPORT COLUMN IS NOT FREE TO EXPAND AS A RESULT OF THE TEMPERATURE RISE BUT IS RESTRAINED BY THE UNHEATED PORTION OF THE COLUMN (3) THE RADIATION IS NEGLECTED, AND (4) THE WALLS ARE VIRTUALLY VERTICAL, THEN THE ABSORPTIVITY MAY BE MODIFIED TO GIVE

$$\bar{\alpha} = 0.5(0.4)$$

$$\bar{\alpha} = 0.20$$

$$\frac{Q_{IN}}{A} = \bar{\alpha}\, \bar{Q} = (0.20)(160)$$

$$\frac{Q_{IN}}{A} = 32 \text{ BTU/HR-FT}^2$$

ASSUMING SOME CONVECTION AND

LETTING $V = 5$ ft/sec

$= 18,000$ FT/HR

$$h_1 = 0.8 + \frac{V}{16,300} = 0.8 + \frac{18,000}{16,300}$$

$$h_1 = 2.0$$

$$h_2 = 2.0$$

$$\frac{Q_{IN}}{A} = \frac{Q_1}{A} + \frac{Q_2}{A}$$

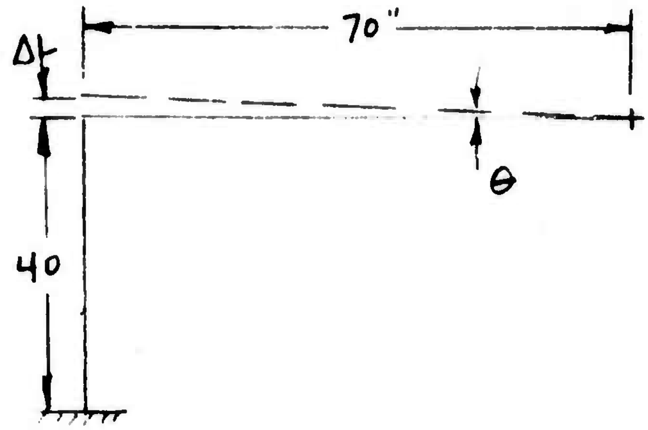
$$\frac{Q_{IN}}{A} = h_1(T_f - T_0) + h_2(T_f - T_0)$$

$$= (h_1 + h_2)(T_f - T_0)$$

$$32 = (2.0 + 2.0)(T_f - T_0)$$

$$T_f - T_0 = 8.0°\ F$$

DETERMINE EFFECTIVE CHANGE IN ELEVATION ANGLE RESULTING FROM 8.0°F TEMPERATURE RISE



$$\Delta L = L \alpha \Delta T$$

$L = 40$ IN

$\alpha = 6 \times 10^{-6} /°F$

$\Delta T = 8.0°F$

$$\Delta L = 40 \times 6 \times 10^{-6} \times 8.0$$

$$\Delta L = 0.00192 \text{ IN}$$

$$\Delta L = 70\theta$$

$$\theta = \frac{0.00192}{70}$$

$$\theta = 27.4 \times 10^{-6} \text{ RAD}$$

$$= 27.4 \times 10^{-6} \times 2.063 \times 10^{5} \text{ SEC} = 5.66 \text{ SEC.}$$

DETERMINE TIME TO REACH TEMPERATURE

$$Q = hA\left(T_f - T\right)$$

$$Q = Wc\,\frac{dT}{dt}$$

$$Wc\,\frac{dT}{dt} = hA\left(T_f - T\right)$$

$h = h_1 + h_2$ (COMBINED FILM COEFFICIENTS OF INNER AND OUTER SURFACE)

c = SPECIFIC HEAT OF SURFACE MATERIAL

$$\frac{dT}{T_f - T} = \frac{hA}{Wc}\,dt$$

LETTING $B = \frac{hA}{Wc}$

$$\int_{T_0}^{T} \frac{dT}{T_f - T} = B \int_0^t dt$$

$$-\ln\left(T_f - T\right)\Big]_{T_0}^{T} = Bt$$

$$\ln \frac{T_f - T}{T_f - T_0} = -Bt$$

T IS TEMPERATURE @ ANY TIME t

$$\frac{T_f - T}{T_f - T_o} = e^{-Bt}$$

$$T_f - T = (T_f - T_o)\, e^{-Bt}$$

$$T - T_f = -(T_f - T_o)\, e^{-Bt}$$

$$T - T_o - (T_f - T_o) = -(T_f - T_o)\, e^{-Bt}$$

$$T - T_o = (T_f - T_o)\left[1 - e^{-\frac{hA}{Wc}t}\right]$$

$h = 4.0$ BTU/FT²-HR

$\frac{W}{A} = 5.2$ LB/FT²

$C = 0.107$ BTU/LB °F

$$\frac{hA}{Wc} = \frac{h}{\frac{W}{A}c} = \frac{4.0}{5.2 \times .107}$$

$$\frac{hA}{Wc} = 7.2$$

TIME CONSTANT

$$\tau = \frac{1}{7.2} = 0.14 \text{ HR}$$

TIME t TO REACH T_f

$$t \approx 5\tau = 5 \times 0.14$$

$$t = 0.70 \text{ HR}$$

APPENDIX B

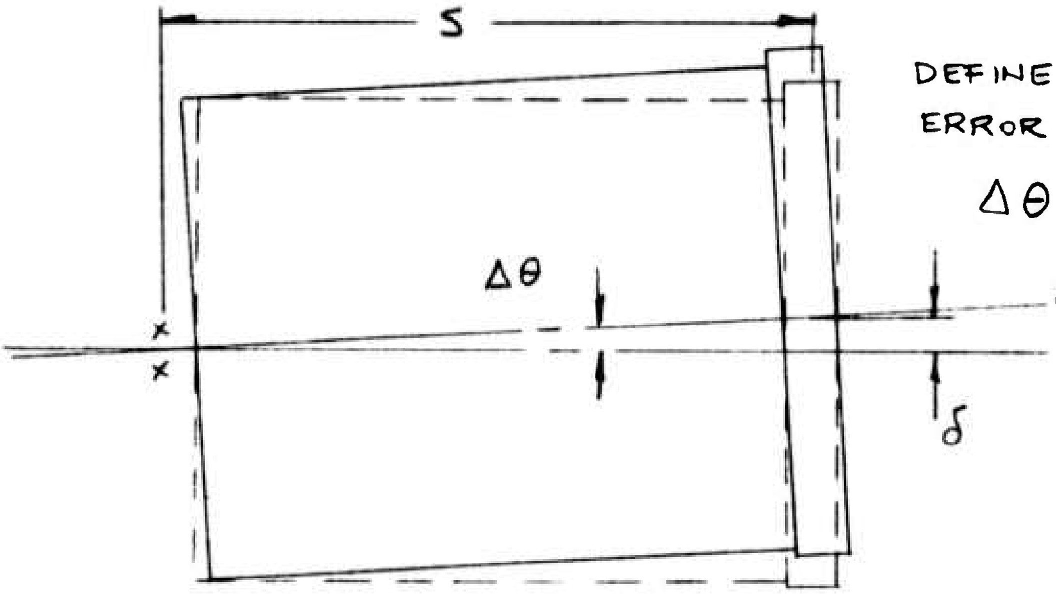
OIL BEARING HEAT POINTING ERROR

THERMAL ANALYSIS

INVESTIGATE THE EFFECT OF VARIATION OF BEARING OIL TEMPERATURE ON COELOSTAT POINTING ERROR.

HEAT FROM OIL PASSING THROUGH THE FLOW RESTRICTORS AND BEARING PADS WILL BE TRANSFERED TO THE ROTOR AND STATOR OF THE BEARING. THIS HEAT WILL IN TURN BE TRANSFERED TO THE STRUCTURE LOCAL TO THE OIL BEARING.

ASSUMING THAT THE STRUCTURE WHICH IS NOT LOCAL TO THE OIL BEARING WILL BE AT AMBIENT TEMPERATURE, AND THAT THE STRUCTURE LOCAL TO THE BEARING WILL BE AT HIGHER TEMPERATURE BECAUSE OF THE HEATED OIL, POINTING ERROR CAN BE EXPECTED AS A RESULT OF DIFFERENTIAL EXPANSION.



DEFINE POINTING ERROR AS

$$\Delta\theta = \frac{\delta}{S}$$

$S = 72''$

ASSUMING FREE EXPANSION OF ELEVATION OIL BEARING COMPONENTS, A 1°F TEMPERATURE DIFFERENCE BETWEEN BEARING AND AMBIENT STRUCTURE REPRESENTS THE FOLLOWING POINTING ERROR

$$\frac{\Delta R}{R} = \alpha \Delta T$$

R = OUTSIDE RADIUS OF BEARING

$\Delta R = \delta$ = CHANGE IN RADIUS (OR ELEVATION)

α = THERMAL COEFF OF EXPANSION – FOR STEEL $\alpha = 6 \times 10^{-6}/°F$

ΔT = MEAN TEMPERATURE DIFFERENCE BETWEEN BEARING RINGS AND SUPPORTING STRUCTURE (OR AMBIENT) NOT LOCAL TO BEARING.

$$\delta = R \alpha \Delta t$$
$$= 30 \times 6 \times 10^{-6} \times 1$$
$$\delta = 0.000180/°F$$

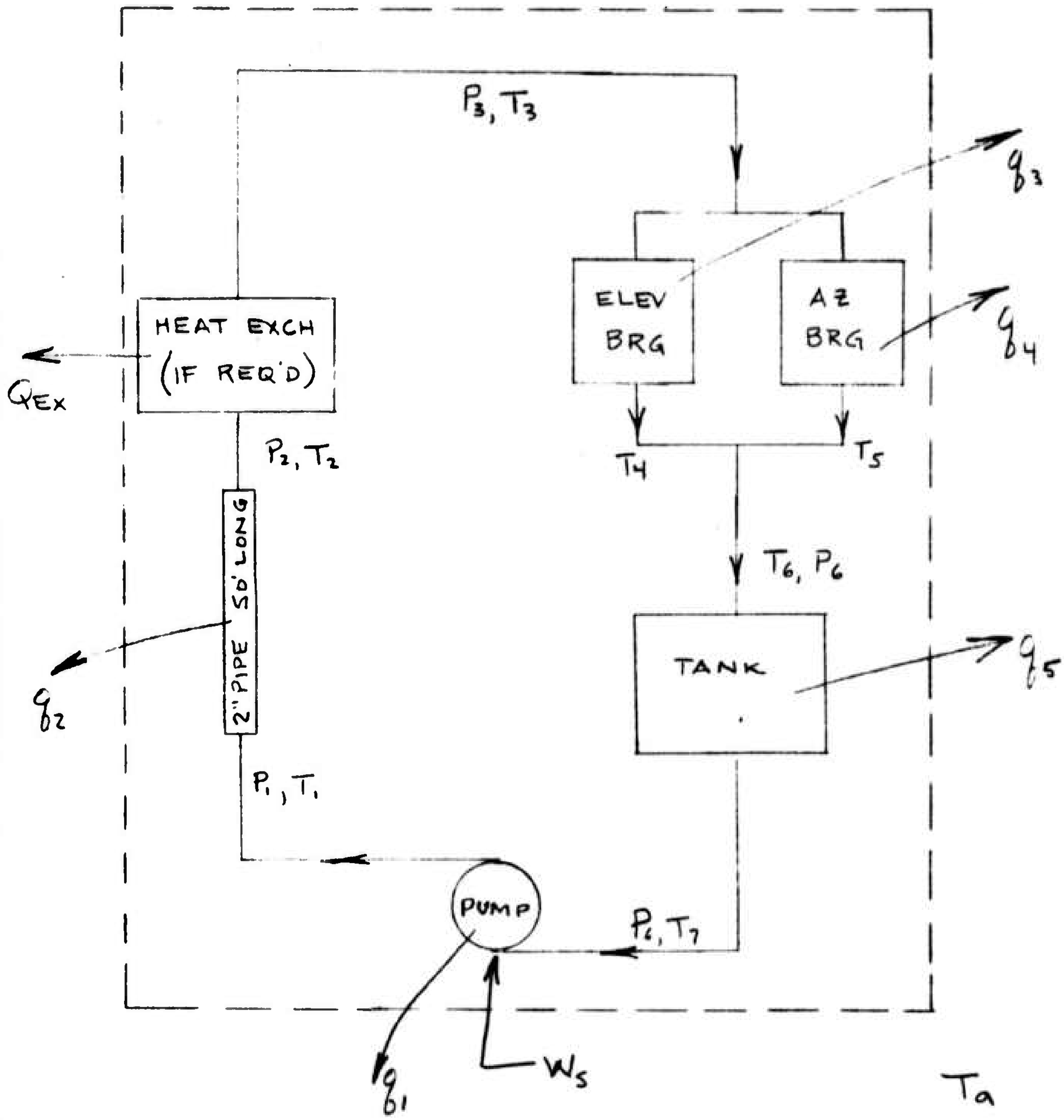
POINTING ERROR PER °F

$$\Delta\theta = \frac{\delta}{S}$$

$$\Delta\theta = \frac{0.00018}{72}$$

$$\Delta\theta = 2.5 \times 10^{-6}\ \text{RAD}/°F$$

CONSIDER THE FOLLOWING SYSTEM



SYSTEM HEAT BALANCE OCCURS WHEN

$$W_s = q_1 + q_2 + Q_{EX} + q_3 + q_4 + q_5 \qquad (1)$$

THE ABOVE APPLIES TO A PER UNIT MASS OF FLOWING FLUID

IN TERMS OF UNIT MASS OF FLOWING FLUID DEFINE THE HEAT FLOWS

(1) FOR 50' LONG 2 IN DIAM PIPE

$$-q_2 = \frac{P_2 - P_1}{\rho} + C_p (T_2 - T_1) \qquad (2)$$

WHERE q_2 = HEAT LEAVING SYSTEM THRU PIPE WALLS (BTU/HR/LB)

T_1 = OIL TEMP AT PIPE ENTRANCE

T_2 = OIL TEMP AT PIPE EXIT

(2) HEAT EXCHANGER

$$-Q_{EX} = \frac{P_3 - P_2}{\rho} + C_p (T_3 - T_2) \qquad (3)$$

(3) ELEVATION BEARING

$$-q_3 = \left[\frac{P_6 - P_3}{\rho} + C_p (T_4 - T_3)\right]\left[F\right] \qquad (4)$$

(4) AZIMUTH BEARING

$$-q_4 = \left[\frac{P_6 - P_3}{\rho} + C_p (T_5 - T_3)\right]\left[1 - F\right] \qquad (5)$$

WHERE (F) IS THAT FRACTION OF FLOW PASSING THRU ELEVATION BEARING

NOTE THAT: $T_6 = f(T_4, T_5, \text{ELEV. FLOW}, \text{AZ. FLOW})$

(5) NEGLECTING ΔP ACROSS TANK

$$-q_5 = C_p (T_7 - T_6) \qquad (6)$$

(6) PUMP

$$-q_1 = -W_s + \frac{P_1 - P_6}{\rho} + C_p (T_1 - T_7) \qquad (7)$$

COMBINE EQ'S (2) THRU (7) INTO EQ (1) TO OBTAIN

$$W_s = \frac{P_1 - P_2}{\rho} + \frac{C_p}{2}\left[2T_7 - T_5 - T_6 + F(T_5 - T_4)\right] \qquad (8)$$

REGARD EQ (1) AS USED UP AND WORK WITH EQ'S (2) THRU (8). CONSIDER q_2, q_5, AND POSSIBLY Q_{EX} (HEAT EXCHANGER OR AIR COOLER MAY OR MAY NOT BE USED DEPENDING ON FINAL TEMPERATURE BALANCE) AS MAJOR HEAT REMOVALS AND REWRITE EQ'S (2) THRU (8)

$$q_2 = \frac{P_1 - P_2}{\rho} + C_p (T_1 - T_2)$$

$$Q_{EX} = \frac{P_2 - P_3}{\rho} + C_p (T_2 - T_3)$$

$$q_3 = \frac{P_6 - P_3}{\rho} + C_p (T_4 - T_3) = 0$$

$$q_4 = \frac{P_6 - P_3}{\rho} + C_p (T_5 - T_3) = 0$$

$$q_5 = C_p (T_6 - T_7)$$

PUMP SHAFT WORK $\quad W_s = + \dfrac{P_6 - P_1}{\rho} + C_p (T_7 - T_1)$

$$W_s = \frac{P_1 - P_2}{\rho} + \frac{C_p}{2} \left[2T_7 - T_5 - T_6 + F(T_5 - T_4) \right]$$

NOTE: $\quad T_5 = T_6 = T_4$

$T_7 = T_1$

REWRITING THE ABOVE EQUATIONS

$$q_2 = \frac{P_1 - P_2}{\rho} + C_p(T_1 - T_2)$$

$$Q_{EX} = \frac{P_2 - P_3}{\rho} + C_p(T_2 - T_3)$$

$$0 = \frac{P_6 - P_3}{\rho} + C_p(T_4 - T_3)$$

$$0 = \frac{P_6 - P_3}{\rho} + C_p(T_4 - T_3)$$

$$q_5 = C_p(T_4 - T_1)$$

PUMP SHAFT WORK FROM FLUID

$$W_s = \frac{P_6 - P_1}{\rho}$$

$$W_s = \frac{P_1 - P_2}{\rho} + \frac{C_p}{2}\left[2T_1 - T_4 - T_4 + F(T_4 - T_4)\right]$$

OVERALL ENERGY BALANCE

$$W_s = q_2 + Q_{EX} + q_5 \qquad (a)$$

HEATS LEAVING FLUID

$$q_2 = \frac{P_1 - P_2}{g} + C_p(T_1 - T_2) \qquad (b)$$

$$Q_{EX} = \frac{P_2 - P_3}{\rho} + C_p(T_2 - T_3) \qquad (c)$$

$$0 = \frac{P_3 - P_6}{\rho} + C_p(T_3 - T_4) \qquad (d)$$

$$q_5 = C_p(T_4 - T_1) \qquad (e)$$

PUMP SHAFT WORK ENTERING FLUID

$$W_s = \frac{P_1 - P_6}{\rho} \qquad (f)$$

LET (m) BE MASS FLOW RATE OF FLOW (LB_m/MIN)

CONVERT EQ'S (b) THRU (e) TO (LB_m/MIN)

$$q_2 = m\frac{P_1 - P_2}{\rho} + m C_p(T_1 - T_2) \quad \text{BTU/MIN} \qquad (g)$$

$$Q_{EX} = m\frac{P_2 - P_3}{\rho} + m C_p(T_2 - T_3) \quad \text{BTU/MIN} \qquad (h)$$

$$0 = (1)\frac{P_3 - P_6}{\rho} + (1) C_p(T_3 - T_4) \quad \text{BTU/MIN} \qquad (i)$$

$$q_5 = m C_p(T_4 - T_1) \quad \text{BTU/MIN} \qquad (j)$$

DEFINE U_2 AS OVERALL HEAT TRANSFER COEFFICIENT FOR HEAT (q_2) UNITS ARE BTU/MIN FT² °F.

SIMILAR DEFINITION FOR OTHER HEATS

$$q_2 = U_2 A_2 \left[\frac{T_1 + T_2}{2} - T_a \right] \quad \text{BTU/MIN} \qquad (k)$$

$$G_{EX} = U_{EX} A_{EX} \left[\frac{T_2 + T_3}{2} - T_a \right] \quad \text{BTU/MIN} \qquad (\ell)$$

$$q_5 = U_5 A_5 \left[\frac{T_4 + T_1}{2} - T_a \right] \quad \text{BTU/MIN} \qquad (m)$$

TEMP T_3 IS AT ENTRANCE TO OIL BEARINGS.

COMBINE EQ'S $(i)(j) + (m)$

$$T_3 = \frac{T_1 \left[1 + \frac{U_5 A_5}{2 m C_p} \right] - T_a \left[\frac{U_5 A_5}{m C_p} \right]}{1 - \frac{U_5 A_5}{2 m C_p}} - \frac{P_3 - P_6}{\rho C_p} \qquad (n)$$

COMBINE EQ'S $(g) + (k)$

$$T_1 = \frac{T_2 \left[1 + \frac{U_2 A_2}{2 m C_p} \right] - T_a \left[\frac{U_2 A_2}{m C_p} \right] + \frac{P_2 - P_1}{\rho C_p}}{1 - \frac{U_2 A_2}{2 m C_p}} \qquad (o)$$

COMBINE EQ'S (h) + (ℓ)

$$T_2 = \frac{T_3\left[1+\frac{U_{EX}A_{EX}}{2mC_p}\right] - T_a\left[\frac{U_{EX}A_{EX}}{mC_p}\right] + \frac{P_3-P_2}{\rho C_p}}{1-\frac{U_{EX}A_{EX}}{2mC_p}} \qquad (p)$$

3 EQUATIONS AND 3 UNKNOWNS. SUBSTITUTE SOME NUMERICAL VALUES AND SOLVE FOR T_3

MAX OIL FLOW

$$m_1 = 65\ IN^3/SEC \times \frac{60\ SEC}{MIN} \times \frac{1}{1728}\frac{FT^3}{IN^3} \times \frac{62.4\ LB}{FT^3} \times .87$$

$$m_1 = 123\ LB/MIN$$

MIN OIL FLOW

$$m_2 = 12\ IN^3/SEC$$

$$m_2 = 22\ LB/MIN$$

$$C_p = 0.50\ BTU/LB\text{-}°F = 4660\ IN\text{-}LB/LB\text{-}°F$$

$$\rho = 0.0322\ LB/IN^3$$

$$A_2 = 26.2\ FT^2$$

$$A_5 = 42.0\ FT^2$$

$U_2 = 0.59$ BTU/MIN FT^2 °F $= 5510$ IN-LB/MIN FT^2 °F

$U_5 = 0.43$ BTU/MIN FT^2 °F $= 4020$ IN-LB/MIN FT^2 °F

$P_1 = 284.7$ PSI

$P_2 = 264.7$ PSI

$P_3 = 244.7$ PSI

$P_6 = 14.7$ PSI

$\rho C_p = 150$ LB/IN^2 °F

$m_1 C_p = 0.574 \times 10^6$ IN-LB/MIN °F

$U_5 A_5 = 0.169 \times 10^6$ IN-LB/MIN °F

$U_2 A_2 = 0.1443 \times 10^6$ IN-LB/MIN °F

$P_3 - P_6 = 230$ PSI

$P_2 - P_1 = -20.0$ PSI

$P_3 - P_2 = -20.0$ PSI

$2 m C_p = 1.15 \times 10^6$ IN-LB/MIN °F

SUBSTITUTE ABOVE NUMERICAL VALUES INTO EQUATIONS (N), (O), + (P).

DETERMINE $T_3 - T_a$

USING VALUES FOR MAXIMUM OIL FLOW

$$T_3 = \frac{T_1\left[1+\dfrac{0.169\times10^6}{1.15\times10^6}\right] - T_a\left[\dfrac{0.169\times10^6}{0.574\times10^6}\right]}{1-\dfrac{0.169\times10^6}{1.15\times10^6}} - \frac{230}{150}$$

$$T_1 = \frac{T_2\left[1+\dfrac{0.1443\times10^6}{1.15\times10^6}\right] - T_a\left[\dfrac{0.1443\times10^6}{0.574\times10^6}\right] - \dfrac{20}{150}}{1-\dfrac{0.1443\times10^6}{1.15\times10^6}}$$

$$T_2 = \frac{T_3\left[1+\dfrac{U_{EX}A_{EX}}{1.15\times10^6}\right] - T_a\left[\dfrac{U_{EX}A_{EX}}{0.574\times10^6}\right] - \dfrac{20}{150}}{1-\dfrac{U_{EX}A_{EX}}{1.15\times10^6}}$$

$$T_3 = \frac{T_1\left[1.147\right] - T_a\left[0.295\right]}{0.853} - 1.53$$

$$T_1 = \frac{T_2\left[1.125\right] - T_a\left[.252\right] - 0.133}{0.875}$$

WITHOUT HEAT EXCHANGER $T_2 = T_3$

THEN SOLVING THE EQUATIONS FOR MAX FLOW

$$T_3 - T_a = 2.36\,^\circ F$$

I.E. FOR THE HIGH FLOW CONDITION THE BEARING INLET TEMP, T_3, SHOULD NOT EXCEED AMBIENT TEMP T_A BY MORE THAN 2.36°F

DETERMINE $T_3 - T_a$
USING VALUES FOR <u>MINIMUM</u> OIL FLOW

$$T_3 = \frac{T_1\left[1 + \frac{0.169 \times 10^6}{0.206 \times 10^6}\right] - T_a\left[\frac{0.169 \times 10^6}{0.103 \times 10^6}\right]}{1 - \frac{0.169 \times 10^6}{0.206 \times 10^6}} - \frac{2.30}{150}$$

$$T_1 = \frac{T_2\left[1 + \frac{0.1443 \times 10^6}{0.206 \times 10^6}\right] - T_a\left[\frac{0.1443 \times 10^6}{0.103 \times 10^6}\right] - \frac{20}{150}}{1 - \frac{0.1443 \times 10^6}{0.206 \times 10^6}}$$

WITHOUT HEAT EXCHANGER $T_2 = T_3$
SOLVING THE EQUATIONS FOR MIN FLOW

$$T_3 - T_a = 0.12\,^\circ F$$

SINCE THE FLOW THROUGH THE BEARING IS DEPENDENT ON VISCOSITY, A CONSTANT VISCOSITY THROUGHOUT THE TEMPERATURE RANGE WOULD MAKE $T_3 - T_a$ CONSTANT. HOWEVER, SINCE THE VISCOSITY, AND HENCE THE FLOW, DOES VARY, $T_3 - T_a$ IS EXPECTED TO VARY WITH IT. (TEMPERATURE VARIATIONS MENTIONED HERE ARE THE OPERATIONAL AMBIENT TEMPERATURE VARIATIONS RANGING FROM -20°F TO 100°F)

THE ANALYSIS IS SOMEWHAT CONSERVATIVE AND ALTHOUGH A TEMP VARIATION, $T_3 - T_a$ IS 2.4°F AT THE HIGHEST FLOW IT IS CONSIDERED HIGH. THE TEMPERATURE DIFFERENCE IS EXPECTED TO BE LESS AND CAN BE DECREASED EVEN FURTHER WITH THE USE OF A HEAT EXCHANGER (OIL COOLER) IF NECESSARY. STRUCTURAL AND PIPING PROVISIONS FOR AN OIL COOLER IN THE SYSTEM HAVE BEEN MADE BUT WILL NOT LIKELY BE REQUIRED.

CONSIDERING A MORE REALISTIC TEMPERATURE RISE OF 1.6°F THE POINTING ERROR WILL LIKELY BE

$$\Delta\theta = 2.5 \times 10^{-6}\ \mathrm{RAD}/{}^\circ\mathrm{F} \times 1.6^\circ\mathrm{F}$$

$$\Delta\theta = 4.0 \times 10^{-6}\ \mathrm{RAD}$$

THE POINTING ERROR CAN HOWEVER BE DIVIDED AND TAKEN ABOUT A MEAN VALUE. RATHER THAN A 4 μRAD POINTING ERROR THE EXCURSION FROM THE MEAN WILL BE ± 2 μRAD

APPENDIX C

AZIMUTH AXIS THRUST BEARING

AZIMUTH AXIS THRUST BEARING

12 SETS OF OPPOSED HYDROSTATIC PADS ARE USED TO PROVIDE THE LOAD CARRYING CAPABILITY OF THE AZIMUTH THRUST BEARING. THE PADS ARE ARRANGED IN A CIRCULAR PATERN HAVING A MEAN RADIUS, r_m, OF 26.88 INCHES, AND ARE EQUALLY SPACED. SUFFICIENT PRELOAD IS PROVIDED TO MEET THE MOMENT STIFFNESS REQUIREMENT OF 19.2×10^9 IN-LB/RAD. CAPILLARY RESTRICTORS ARE USED AS THE FLOW COMPENSATING ELEMENTS SO THAT PROPER DISTRIBUTION TO EACH PAD IS ASSURED. ONE CAPILLARY RESTRICTOR PER PAD IS USED.

LIST OF SYMBOLS

K_{M_R} = REQUIRED ROCKING MOMENT STIFFNESS OF BEARING (IN-LB/RAD)

K_M = ACTUAL ROCKING MOMENT STIFFNESS OF BEARING (IN-LB/RAD)

M_{xx} = ROCKING MOMENT (IN-LB)

ψ = ANGULAR TILT CAUSED BY M_{xx} (RAD)

K_0 = STIFFNESS OF SINGLE PAIR OF OPPOSED PADS (LB/IN)

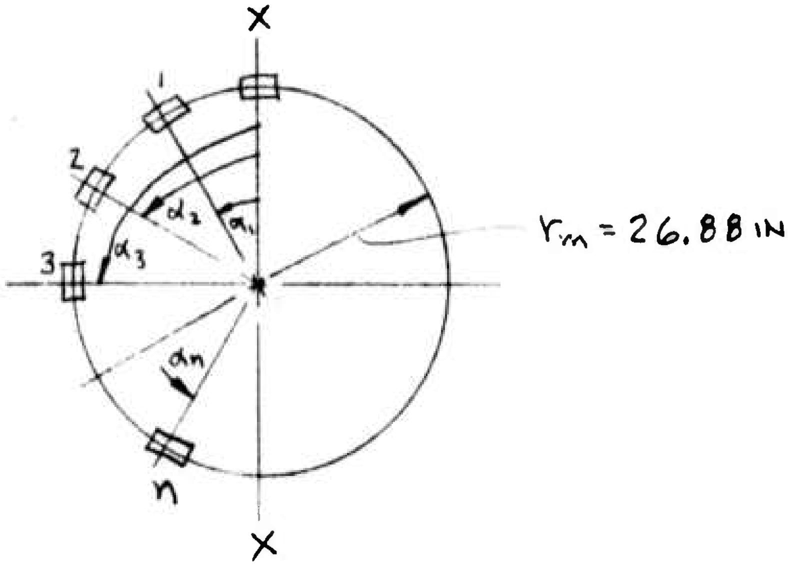
S_1 = STIFFNESS OF INDIVIDUAL UPPER PAD (LB/IN)

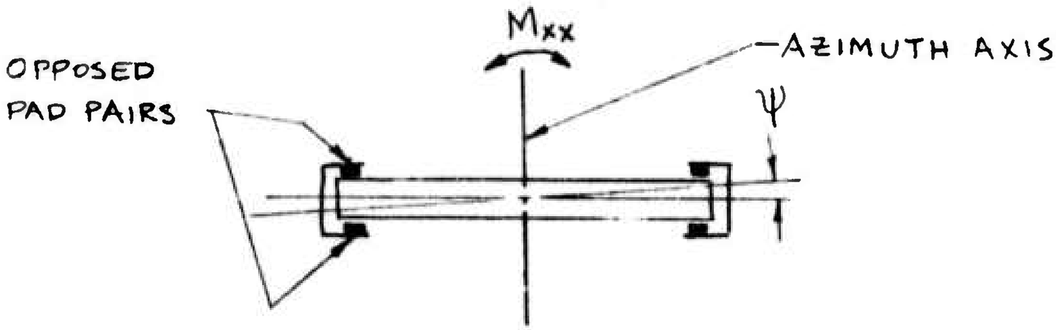
S_2 = STIFFNESS OF INDIVIDUAL LOWER PAD (LB/IN)

r_m = MEAN RADIUS OF PAD CIRCLE (IN)

α = ANGULAR LOCATION OF PADS FROM X-X REF (DEG)

BEARING PAD ARRANGEMENT





ROCKING MOMENT

$$M_{xx} = \sum_{n=1}^{12} K_0 r_m^2 \sin \alpha_n \psi$$

SUBSTITUTING VALUES FOR 12 EQUALLY SPACED PAD PAIRS

$$M_{xx} = 2 K_0 r_m^2 \psi \left[2 \sin 30^\circ + 2 \sin 60^\circ + \sin 90^\circ \right]$$

$$M_{xx} = 7.44 K_0 r_m^2 \psi$$

MOMENT STIFFNESS

$$K_m = \frac{M_{xx}}{\psi} = 7.44\, K_o r_m^2$$

REQUIRED BEARING STIFFNESS

$$K_{M_R} = 19.2 \times 10^9 \text{ IN-LB/RAD}$$

REQUIRED PAD PAIR STIFFNESS

$$K_{O_R} = \frac{K_{M_R}}{7.44\, r_m^2}$$

$$K_{O_R} = \frac{19.2 \times 10^9}{7.44 \times 26.88^2}$$

$$K_{O_R} = 3.58 \times 10^6 \text{ LB/IN}$$

INCLUDE 20% SAFETY MARGIN

USE $K_{O_R} = 4.4 \times 10^6$ LB/IN FOR BEARING SIZING

THRUST PAD DETERMINATION

LIST OF SYMBOLS

β = PRESSURE RATIO

P_r = RECESS PRESSURE (PSI)

P_s = SUPPLY PRESSURE (PSI)

S_1 = STIFFNESS OF UPPER PAD (LB/IN)

S_2 = STIFFNESS OF LOWER PAD (LB/IN)

W = PAD LOAD (LB)

h = FILM THICKNESS (INCH)

C = AXIAL CLEARANCE (INCH)

μ = VISCOSITY (REYN)

A_p = PAD AREA (IN^2)

a_f = PAD AREA COEFF.

$A_e = a_f A_p$ = EFFECTIVE PAD AREA (IN^2)

Q = PAD FLOW (IN^3/SEC)

q_f = PAD FLOW COEFF.

e = ECCENTRICITY (INCH)

ϵ = ECCENTRICITY RATIO $\left(\frac{e}{c}\right)$

d_c = CAPILLARY DIAMETER

l_c = CAPILLARY LENGTH

NOTES (1) ALL PRIMED SYMBOLS(') DENOTE PRELOAD CONDITION

(2) REFERENCE DESIGN MANUAL IS "CAST BRONZE HYDROSTATIC BEARING DESIGN" BY HARRY RIPPEL OF FRANKLIN INSTITUTE

DETERMINE LOAD DISPLACEMEND CHARACTERISTICS FOR 2 PAD OPPOSED BEARING (CAPILLARY COMPENSATION)

$$Z = \frac{R}{a_f A_p P_s} = \frac{\beta'}{1-\beta'}\left[\frac{1}{(1-\epsilon)^3 + \left(\frac{\beta'}{1-\beta'}\right)} - \frac{1}{(1+\epsilon)^3 + \left(\frac{\beta'}{1-\beta'}\right)}\right]$$

FOR A PRELOAD PRESSURE RATIO

$$\beta' = 0.5$$

PLOT $\frac{R}{a_f A_p P_s}$ VS ϵ

WHERE R = APPLIED LOAD ON OPPOSED BEARING PAD

FOR $\epsilon = 0.1$

$$Z = \frac{0.5}{1-0.5}\left[\frac{1}{1.741} - \frac{1}{2.33}\right] = 0.145$$

FOR $\epsilon = 0.2$

$$Z = \left[\frac{1}{1.512} - \frac{1}{2.726}\right] = 0.295$$

FOR $\epsilon = 0.3$

$$Z = \left[\frac{1}{1.343} - \frac{1}{3.195}\right] = 0.431$$

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ENGINEERING DATA

LOAD DISPLACEMENT CURVE FOR
2 PAD OPPOSED BEARING

$\frac{R}{a_f A_p P_s}$ vs ϵ

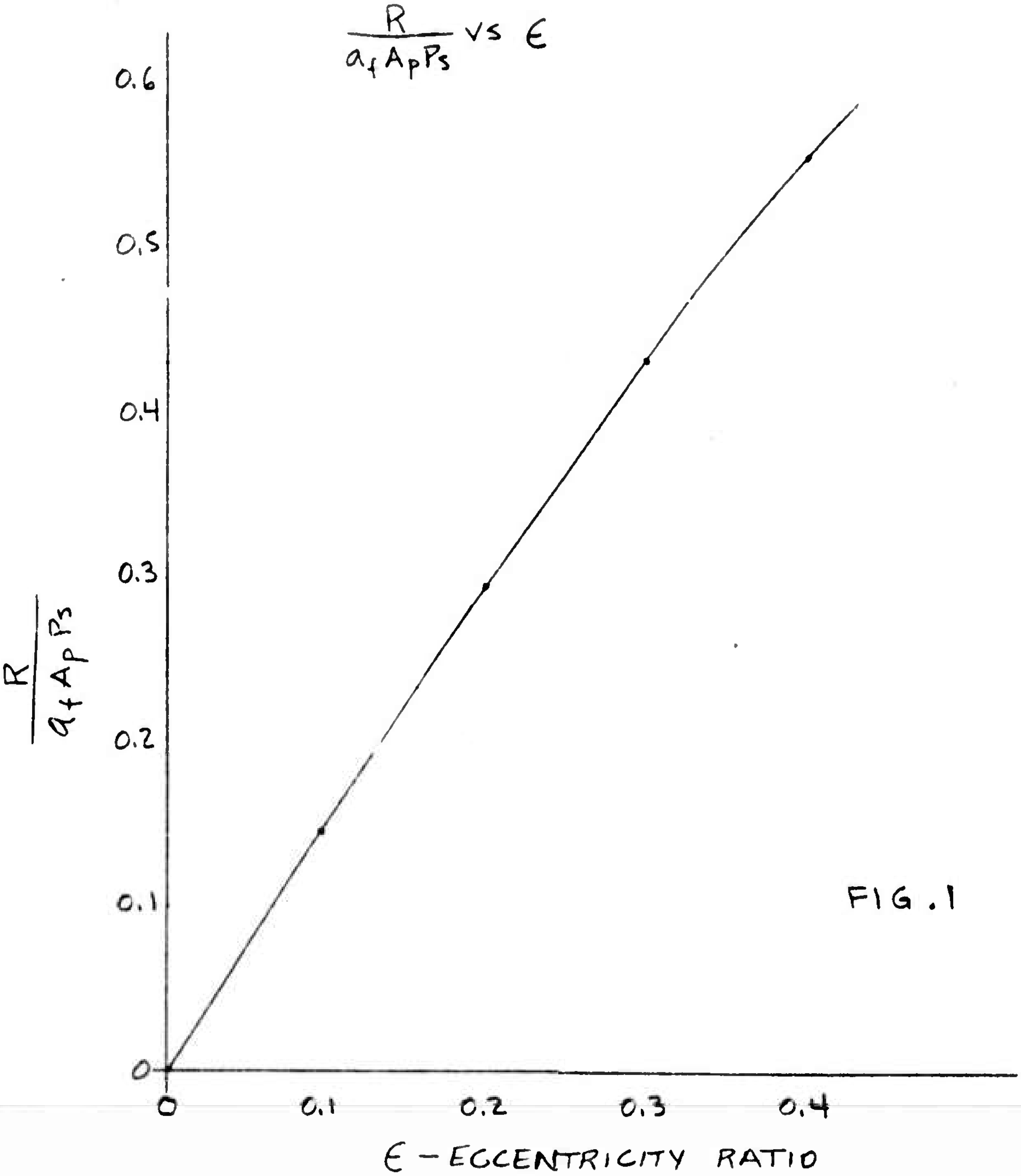

FIG. 1

SUBJECT | PAGE
BY DATE

SEVERAL ITERATIONS WERE DONE TO ARRIVE AT THE FINAL OPTIMIZED PAD CONFIGURATION AND SUPPLY PRESSURE. ONLY THE FINAL ITERATION IS SHOWN.

APPLIED LOAD PER PAD, R

$$R = \frac{\text{SUPPORTED WEIGHT}}{12} = \frac{39{,}000}{12}$$

$$R = 3250 \text{ LBS.}$$

PRELIMINARY ECCENTRICITY ESTIMATE

TREATING THE BEARING AS A SPRING

$$\text{DISPL (ECCENTRICITY) } e = \frac{R}{K_{OR}}$$

$$e = \frac{3250}{4.4 \times 10^{6}}$$

$$e = 0.74 \times 10^{-3} \text{ INCH}$$

BEARING AXIAL CLEARANCE $C = 0.0025$

$$\text{ECCEN RATIO } \epsilon = \frac{e}{C} = \frac{0.74 \times 10^{-3}}{2.5 \times 10^{-3}}$$

$$\epsilon = 0.296 \approx 0.3$$

REFERING TO THE STIFFNESS CURVE FOR $\epsilon = 0.3$

$$\frac{R}{a_f A_p P_s} = 0.431$$

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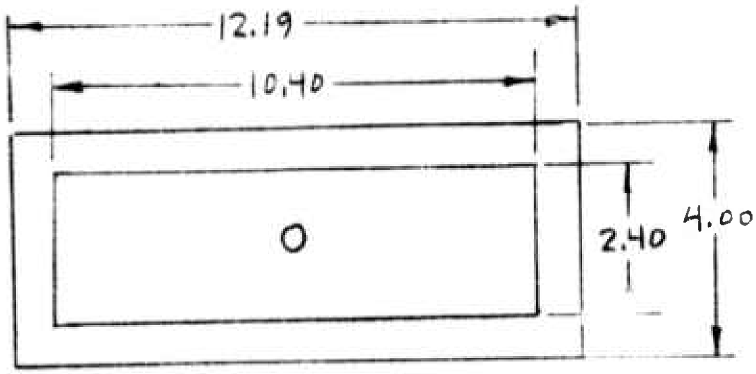
ENGINEERING DATA

FINAL PAD CONFIGURATION



AREA OF PAD

$$A_p = 4 \times 12.19$$

$$A_p = 48.7 \text{ IN}^2$$

DETERMINE PAD COEFFICIENTS FOR AREA AND FLOW

$$\frac{y}{Y} = \frac{2.40}{4.00} = 0.60$$

$$\frac{X}{Y} = \frac{12.19}{4.00} = 3.04$$

FROM REF. DESIGN MANUAL BY RIPPEL

$$a_f = 0.743$$

$$q_f = 3.90$$

SUBJECT

BY DATE

PAGE

SUPPLY PRESS

REMEMBERING $\frac{R}{a_f A_p P_s} = 0.431$ FOR $\epsilon = 0.3$

DETERMINE SUPPLY PRESS

$$a_f A_p = (0.743)(48.7) = 36.2 \text{ IN}^2$$

$$R = 3250 \text{ LB}$$

$$P_s = \frac{3250}{(0.431)(36.2)}$$

$$P_s = 208 \text{ psi}$$

ALLOW 10% SAFETY MARGIN

USE $P_s = 230$ PSI

ACTUAL ECCENTRICITY AT $P_s = 230$ PSI

$$\frac{R}{a_f A_p P_s} = \frac{3250}{(36.2)(230)} = 0.39$$

REFERING TO THE GRAPH IN FIG. 1

$$\epsilon = 0.27$$

$$e = \epsilon C = (0.27)(0.0025)$$

$$e = 0.000675 \text{ IN}$$

PAD PRELOAD

$$\beta' = \frac{P_r'}{P_s} = 0.50$$

$$P_r' = 0.50 \times 230$$

$$P_r' = 115 \text{ PSI}$$

$$W' = a_f A_p P_r' = (36.2)(115)$$

$$W' = 4160$$

PAD FLOWS

THE FLOW THRU THE BEARING PAD CLEARANCE IS LAMINAR HENCE IT VARIES INVERSELY WITH VISCOSITY

FLOW PER PAD $Q' = q_f a_f P_r' \left(\frac{h'^3}{\mu}\right)$

MINIMUM EXPECTED VISCOSITY $\mu = 4 \times 10^{-6}$ REYN

$$q_f = 3.90$$

$$a_f = 0.743$$

$$P_r' = 115 \text{ PSI}$$

$h_1' = 0.0025$ IN (@ PRELOAD-NO ECCENTRICITY)

$$Q' = (3.90)(0.743)(115)\frac{(0.0025)^3}{4 \times 10^{-6}}$$

$$Q' = 1.30 \text{ IN}^3/\text{SEC}$$

LOAD FLOWS

$$\frac{Q_n}{Q'} = \frac{(1-\epsilon\cos\theta_n)^3\left(\frac{1}{\beta'}\right)}{\left[(1-\epsilon\cos\theta_n)^3\left(\frac{1-\beta'}{\beta'}\right)+1\right]}$$

SUBSCRIPTS

1 - UPPER PAD
2 - LOWER PAD

$$\frac{Q_1}{Q'} = \frac{(1+0.27)^3\left(\frac{1}{0.5}\right)}{\left[(1+0.27)^3\left(\frac{1-0.5}{0.5}\right)+1\right]}$$

$$\frac{Q_1}{Q'} = \frac{4.10}{3.05} = 1.342$$

$$Q_1 = 1.342 \times 1.30$$

$$Q_1 = 1.75 \text{ IN}^3/\text{SEC}$$

$$\frac{Q_2}{Q'} = \frac{(1-0.27)^3(2)}{\left[(1-0.27)^3(1)+1\right]}$$

$$\frac{Q_2}{Q'} = \frac{0.778}{1.389} = 0.56$$

$$Q_2 = 0.56 \times 1.30$$

$$Q_2 = 0.729 \text{ IN}^3/\text{SEC}$$

TOTAL BEARING FLOW – 24 PADS (12 UPPER AND 12 LOWER)

$$Q_T = 12\,Q_1 + 12\,Q_2$$

$$= 12(1.75) + 12(0.73)$$

$$= 12(2.48)$$

$$Q_T = 29.8 \text{ IN}^3/\text{SEC} \quad (7.75\ \text{GPM})$$

FLOW HORSEPOWER

$$H_P = \frac{29.8 \times 230}{6600}$$

$$H_P = 1.04\ hp$$

NOTE: THE ABOVE FLOWS ARE BASED ON THE MINIMUM VISCOSITY OF THE BEARING FLUID. DURING COLD WEATHER OPERATION (NOV. 15 - APRIL 30) THE OPERATIONAL FLUID WILL BE MIL-H-5606B (HUMBLE UNIVIS J41).

DURING WARM WEATHER OPERATION (MAY 1, – NOV. 14) THE OPERATIONAL FLUID WILL BE AUTOMATIC TRANSMISSION FLUID (EXXON GLIDE OR EQUIVALENT). THE TWO FLUIDS – COLD AND WARM WEATHER – ARE PETROLEUM BASED AND ARE COMPATABLE. HOWEVER, DURING CHANGEOVER THE SYSTEM SHOULD BE DRAINED AS WELL AS POSSIBLE BEFORE ADDING THE NEW FLUID. THIS WILL PREVENT THE FORMATION OF A THIRD FLUID, WHICH WILL BE A MIXTURE AND WHOSE VISCOSITY CHARACTERISTICS WILL BE DOUBTFUL. IDEALLY NO VARIATION IN VISCOSITY SHOULD OCCUR. REALISTICALLY, HOWEVER, IT IS DESIREABLE TO MAINTAIN THE VISCOSITY OF THE FLUID BETWEEN THE LIMITS OF 4×10^{-6} TO 28×10^{-6} REYN (32 TO 200 CENTISTOKES)

CHECK FINAL BEARING STIFFNESS

STIFFNESS OF ONE OPPOSED PAIR OF PADS, K_0, IS EQUAL TO THE SUM OF THE STIFFNESSES OF THE OPPOSING PADS.

$$K_0 = S_1 + S_2$$

S_1 = STIFFNESS OF UPPER PAD

S_2 = STIFFNESS OF LOWER PAD

$$S_n = \frac{3W_n}{h_n}\left(1-\beta_n\right)$$

W, β, AND h MUST BE DETERMINED

FILM THICKNESS

$$h_n = C + e_n$$

$$e = \epsilon C$$

$$\epsilon = 0.27$$

$$C = 0.0025$$

$$e = (0.27)(0.0025) = 0.000675$$

$$h_1 = 0.0025 + 0.000675 = 0.003175$$

$$h_2 = 0.0025 - 0.000675 = 0.001825$$

PAD LOAD $$W_n = \frac{A_p \mu Q_n}{q_f h_n}$$

$$W_1 = \frac{(48.7)(4\times10^{-6})(1.75)}{(3.9)(0.003175)^3} = 2730 \text{ LB}$$

$$W_2 = \frac{(48.7)(4\times10^{-6})(0.729)}{(3.9)(0.001825)^3} = 5970 \text{ LB}$$

QUICK CHECK

$$W_2 - W_1 = 5970 - 2730 = 3240 \approx R \quad \text{OK}$$

PAD PRESSURE RATIO β_n

$$P_{r_1} = \frac{W_1}{A_p a_f} = \frac{2730}{36.2} = 75.5 \text{ PSI}$$

$$P_{r_2} = \frac{5970}{36.2} = 165 \text{ PSI}$$

$$\beta_1 = \frac{P_{r_1}}{P_s} = \frac{75.5}{230} = 0.328$$

$$\beta_2 = \frac{P_{r_2}}{P_s} = \frac{165}{230} = 0.717$$

STIFFNESS $S_1 = \frac{3 W_1}{h_1}(1-\beta_1)$

$$S_1 = \frac{3(2730)}{0.003175}(1-0.328) = 1.73 \times 10^6 \text{ LB/IN}$$

$$S_2 = \frac{3(5970)}{0.001825}(1-0.717) = 2.78 \times 10^6 \text{ LB/IN}$$

$$K_0 = S_1 + S_2$$

$$= 1.73 \times 10^6 + 2.78 \times 10^6 \text{ LB/IN}$$

$$K_0 = 4.51 \times 10^6 \text{ LB/IN}$$

MARGIN OF SAFETY

$$m = \frac{K_0 - K_{0R}}{K_{0R}} = \frac{4.51 - 3.58}{3.58} = 0.26$$

$$m = 26\%$$

SIZING OF CAPILLARY RESTRICTORS

EQUATE PAD FLOW TO CAPILLARY FLOW

$$Q' = q_f a_f P_r' \frac{h'^3}{\mu} = \frac{\pi d_c^4 (P_s - P_r')}{128 \mu l_c}$$

WHERE d_c = CAPILLARY TUBE DIAMETER

l_c = CAPILLARY TUBE LENGTH

$$\beta' = \frac{P_r'}{P_s} = 0.5$$

$$\frac{d_c^4}{l_c} = \left(\frac{\beta'}{1-\beta'}\right) \cdot \frac{128}{\pi} a_f q_f h'^3$$

$$\frac{d_c^4}{l_c} = \left(\frac{0.5}{1-0.5}\right)\left(\frac{128}{\pi}\right)(0.743)(3.9)(0.0025)^3$$

$$\frac{d_c^4}{l_c} = 184 \times 10^{-8}$$

USING STANDARD TUBING $d_c = 0.054$ IN

$$l_c = \frac{(0.054)^4}{184 \times 10^{-8}} = \frac{852}{184}$$

$$l_c = 4.63 \text{ IN}$$

CHECK MAXIMUM REYNOLS NUMBER

Q_{MAX} OCCURS AT $\mu = 4 \times 10^{-6}$ REYN

$$N_R = \frac{4 \rho Q_c}{\pi d_c \mu}$$

$$Q_{MAX} = 1.75 \text{ IN}^3/\text{SEC}$$

$$N_R = \frac{\cancel{4} \times 89.3 \times \cancel{10^{-6}} \times 1.75}{\pi \times 0.054 \times \cancel{4 \times 10^{-6}}}$$

$N_R = 920 < 2000$ OK

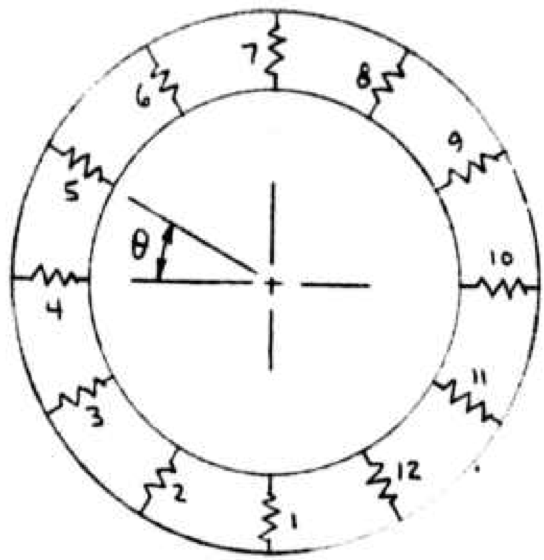
APPENDIX D

AZIMUTH AXIS RADIAL BEARING

AZIMUTH AXIS RADIAL BEARING

THE AZIMUTH AXIS RADIAL BEARING IS A MULTIPLE PAD JOURNAL BEARING HAVING 12 EQUALLY SPACED PADS. THE BEARING IS 58.00 INCHES IN DIAMETER AND PROVIDES LATERAL RESTRAINT OF THE AZIMUTH AXIS SO THAT NO ECCENTRICITY OF THE AXIS CAN OCCUR.

BEARING PAD ARRANGEMENT



PAD STIFFNESS $S_1 = S_2 = S_3 = S_n = S$

REQUIRED BEARING LATERAL STIFFNESS $K_{T_R} = 6 \times 10^6$ LB/IN

OWENS-ILLINOIS
FECKER SYSTEMS DIVISION

ENGINEERING DATA

ACTUAL STIFFNESS $K_T = 2S + 4S \sin^2 60° + 4S \sin^2 30$

$K_T = 6S = 6 \times 10^6$

REQUIRED PAD STIFFNESS $S_R = 1 \times 10^6$ LB/IN

SIZE BEARING PADS USING VALUE
FOR BEARING STIFFNESS $S = 1 \times 10^6$ LB/IN

SUBJECT

BY DATE

PAGE

FORM E 106

RADIAL PAD DETERMINATION

LIST OF SYMBOLS

β = PRESSURE RATIO
P_r = RECESS PRESSURE (PSI)
P_s = SUPPLY PRESSURE (PSI)
S = PAD STIFFNESS (LB/IN)
W = PAD LOAD (LB)
h = FILM THICKNESS (INCH)
C = RADIAL CLEARANCE (INCH)
μ = VISCOSITY (REYN)
A_p = PAD AREA (IN^2)
a_f = PAD AREA COEFFICIENT
Q = PAD FLOW (IN^3/SEC.)
q_f = PAD FLOW COEFFICIENT
e = ECCENTRICITY (INCH)
$\epsilon = e/c$ = ECCENTRICITY RATIO
d_c = CAPILLARY TUBE DIAMETER (IN)
l_c = CAPILLARY TUBE LENGTH (IN)

NOTES (1) ALL PRIMED SYMBOLS (') DENOTE PRELOAD CONDITION

(2) REFERENCE DESIGN MANUAL IS "CAST BRONZE HYDROSTATIC BEARING DESIGN" BY HARRY RIPPEL OF FRANKLIN INSTITUTE.

SINCE NO GREATER THAN NEGLIGIBLE ECCENTRICITY WILL EVER BE REALIZED BY THE JOURNAL ONLY THE PRELOAD CONDITIONS NEED BE CONSIDERED.

USING CAPILLARY COMPENSATION

$$S' = \frac{3W'}{h'}(1-\beta')$$

$$W' = \frac{S'h'}{3(1-\beta')}$$

$$h' = C = 0.0025 \text{ IN}$$

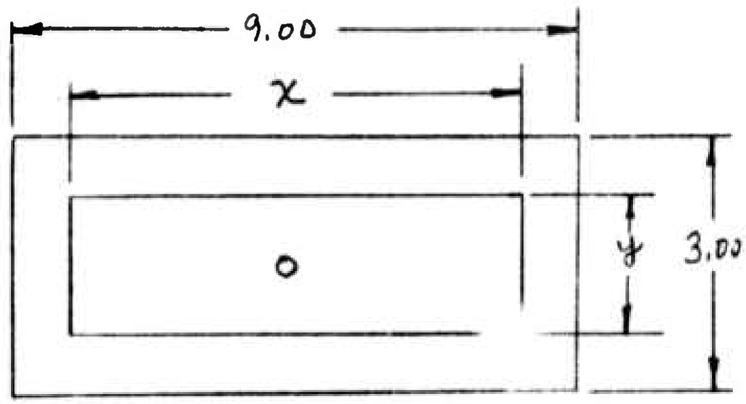
$$S = 1 \times 10^6 \text{ LB/IN}$$

AND USING $\beta' = 0.3$

$$W' = \frac{(1\times10^6)(0.0025)}{3(1-0.3)}$$

$$W' = 1192 \text{ LB}$$

USING CONVENIENT PAD CONFIGURATION



PAD AREA

$$A_p = XY = 9.0 \times 3.0$$

$$A_p = 27.0 \text{ IN}^2$$

$$W' = a_f A_p P_r' = a_f A_p \beta' P_s$$

USING AVAILABLE SUPPLY PRESS $P_s = 230$ psi

$$a_f = \frac{W'}{A_p \beta' P_s} = \frac{1192}{(27.0)(0.3)(230)}$$

PAD AREA COEFFICIENT

$$a_f = 0.64$$

DETERMINE INSIDE DIMENSIONS x AND y

$$\frac{X}{Y} = \frac{9.00}{3.00} = 3.0$$

FROM REF. DESIGN MANUAL BY RIPPEL

$$\frac{y}{Y} = 0.44$$

$$y = (0.44)(3.00)$$

$$y = 1.32 \text{ IN}$$

$$Y - y = X - x$$

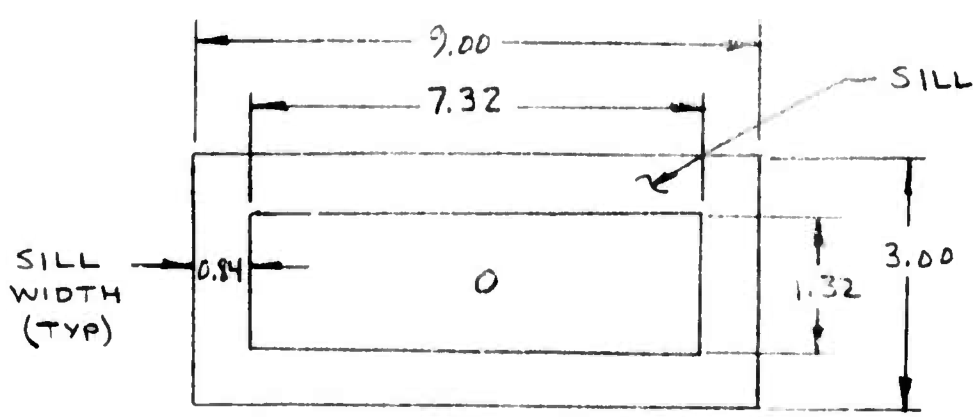
$$x = X - (Y - y) = 9.00 - (3.00 - 1.32)$$

$$x = 7.32 \text{ IN}$$

SILL WIDTH $L = \frac{Y - y}{2} = \frac{3.00 - 1.32}{2}$

$$L = 0.84 \text{ IN}$$

FINAL PAD CONFIGURATION



PAD FLOW

THE PAD FLOW THRU THE BEARING PAD CLEARANCE IS LAMINAR HENCE IT VARIES INVERSELY WITH VISCOSITY

FROM REF. DESIGN MANUAL BY RIPPEL

FOR $\frac{X}{Y} = 3.0$ AT $\frac{y}{Y} = 0.44$

$$q_f = 2.95$$

$$Q' = q_f \frac{W' h'^3}{A_p \mu}$$

MINIMUM EXPECTED VISCOCITY $\mu = 4 \times 10^{-6}$ REYN

$q_f = 2.95$

$W' = 1192$ LB

$h'^3 = 0.0025$

$A_p = 27.0$ IN^2

$$Q' = 2.95 \times \frac{1192}{27.0} \times \frac{0.0025^3}{4 \times 10^{-6}}$$

$$Q' = 0.51 \text{ IN}^3/\text{SEC}$$

TOTAL FLOW - 12 PADS

$$Q_T = 12Q' = 12 \times 0.51$$

$$Q_T = 6.12 \text{ IN}^3/\text{SEC} \quad (1.6 \text{ GPM})$$

FLOW HORSEPOWER

$$H_p = \frac{(6.12)(230)}{6600} = 0.214 \text{ hp}$$

SIZING OF CAPILLARY

EQUATE PAD FLOW TO CAPILLARY FLOW

$$Q' = q_f a_f P_r' \frac{h'^3}{\mu} = \frac{\pi d_c^4 (P_s - P_r')}{128 \mu l_c}$$

WHERE d_c = CAPILLARY TUBE DIAMETER

l_c = CAPILLARY TUBE LENGTH

$$\beta' = \frac{P_r'}{P_s} = 0.3$$

$$\frac{d_c^4}{l_c} = \frac{\beta'}{1-\beta} \quad \frac{128}{\pi} a_f g_f h'^3$$

$$\frac{d^4}{l_c} = \left(\frac{0.3}{1-0.3}\right)\left(\frac{128}{\pi}\right)(0.64)(2.95)(0.0025)^3$$

$$\frac{d_c^4}{l_c} = 51.5 \times 10^{-8}$$

USING STANDARD TUBING $d_c = 0.035$

$$l_c = \frac{d_c^4}{51.5 \times 10^{-8}} = \frac{150 \times 10^{-8}}{51.5 \times 10^{-8}}$$

CAPILLARY LENGTH $l_c = 2.91$ INCH

CHECK MAXIMUM REYNOLDS NUMBER

$$N_R = \frac{4 \rho Q_c}{\pi d_c \mu}$$

$$N_R = \frac{4 \times 89.3 \times 10^{-6} \times 0.51}{\pi (0.035)(4 \times 10^{-6})}$$

$$N_R = 412 < 2000 \qquad \text{OK}$$

APPENDIX E

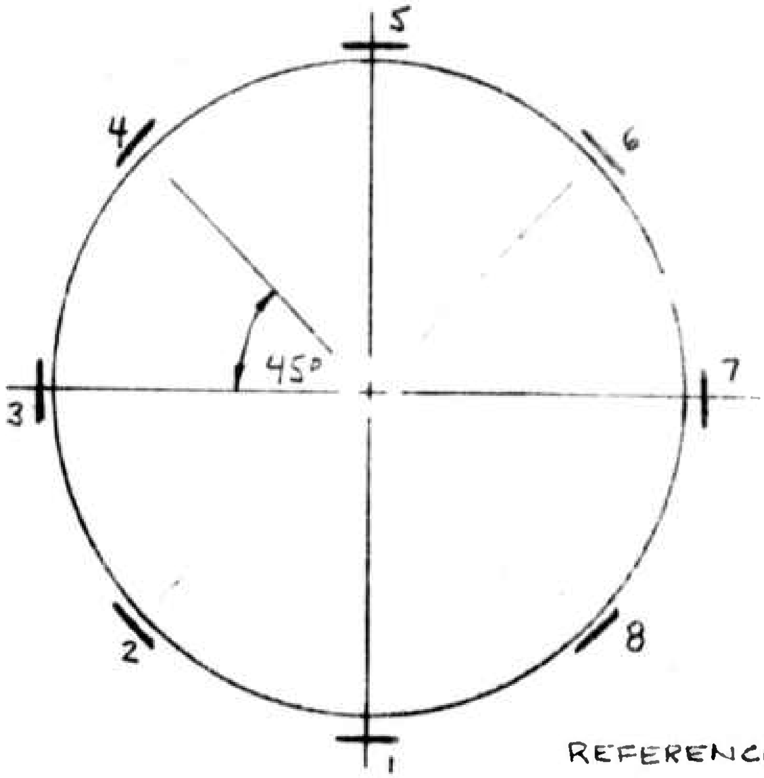
ELEVATION AXIS RADIAL BEARING

ELEVATION AXIS RADIAL BEARING

THE ELEVATION AXIS RADIAL BEARING IS A JOURNAL BEARING WHICH SUPPORTS THE ELEVATION AXIS AT THE INBOARD LOCATION SO THAT ITS AXIS IS HORIZONTAL. THE BEARING CONSISTS OF 8 EQUALLY SPACED HYDROSTATIC PADS WHICH ARE PRELOADED TO PROVIDE A RADIAL STIFFNESS OF APPROXIMATELY 5.5×10^6 LB/IN. THIS IS SOMEWHAT GREATER THAN THE REQUIRED STIFFNESS OF 3.0×10^6 LB/IN DUE PRIMARILY TO SMALLER JOURNAL ECCENTRICITY WHICH IS USED FOR GREATER STABILITY.

CAPILLARY RESTRICTORS ARE USED AS THE FLOW COMPENSATING ELEMENTS SO THAT PROPER FLUID DISTRIBUTION TO EACH PAD IS ASSURED. ONE CAPILLARY RESTRICTOR FOR EACH PAD IS USED. BEARING DIAMETER IS 55.00 IN.

8 PAD JOURNAL BEARING



REFERENCE PAD IS #1 PAD

$\theta_1 = 0°$	$\cos\theta_1 = 1.0$
$\theta_2 = 45°$	$\cos\theta_2 = 0.707$
$\theta_3 = 90°$	$\cos\theta_3 = 0$
$\theta_4 = 135°$	$\cos\theta_4 = -0.707$
$\theta_5 = 180°$	$\cos\theta_5 = -1.0$
$\theta_6 = 225°$	$\cos\theta_6 = -0.707$
$\theta_7 = 270$	$\cos\theta_7 = 0$
$\theta_8 = 315°$	$\cos\theta_8 = 0.707$

LIST OF SYMBOLS

β = PRESSURE RATIO
P_r = RECESS PRESSURE (PSI)
P_s = SUPPLY PRESSURE (PSI)
S = INDIVIDUAL PAD STIFFNESS (LB/IN)
W = PAD LOAD
h = FILM THICKNESS (IN)
C = RADIAL CLEARANCE (IN)
e = ECCENTRICITY (IN)
$\epsilon = e/c$ = ECCENTRICITY RATIO
μ = VISCOSITY (REYN)
A_p = PAD AREA (IN^2)
a_f = PAD AREA COEFFICIENT
Q = PAD FLOW (IN^3/SEC)
q_f = PAD FLOW COEFFICIENT
d_c = CAPILLARY TUBE DIAMETER (IN)
l_c = CAPILLARY TUBE LENGTH (IN).

NOTES (1) ALL PRIMED SYMBOLS (') DENOTE PAD PRELOAD CONDITION

(2) REFENCE DESIGN MANUAL IS "CAST BRONZE HYDROSTATIC BEARING DESIGN" BY HARRY RIPPEL OF FRANKLIN INSTITUTE.

DETERMINE LOAD DISPLACEMENT CHARACTERISTICS FAR A MULTIPAD JOURNAL BEARING HAVING EIGHT (8) EQUALLY SPACED PADS.

FROM REF DESIGN MANUAL BY H. RIPPEL

$$\frac{R}{a_f A_p P_s} = \frac{\beta'}{1-\beta'}\left[\frac{\cos\theta_1}{(1-\epsilon\cos\theta_1)^3+\left(\frac{\beta'}{1-\beta'}\right)} + \cdots\cdots \frac{\cos\theta_n}{(1-\epsilon\cos\theta_n)^3+\left(\frac{\beta'}{1-\beta'}\right)}\right]$$

PLOT LOAD-DISPLACEMENT CHARACTERISTICS I.E.

$$\frac{R}{a_f A_p P_s} \text{ V } \epsilon$$

WHERE R = BEARING LOAD

ϵ = ECCENTRICITY RATIO

SEVERAL ITERATIONS WERE PERFORMED TO OBTAIN A DESIREABLE PAD CONFIGURATION. ONLY THE FINAL ITERATION IS INCLUDED.

USE $\beta' = 0.15$

OBTAIN VALUES OF $\frac{R}{a_f A_p P_s}$ FOR

ECCENTRICITIES OF 0.1, 0.2, 0.3, AND 0.4 AND PLOT

$$\frac{\beta'}{1-\beta'} = \frac{0.15}{1-0.15} = 0.1765$$

FOR $\epsilon = 0.1$

$$\frac{R}{a_f A_p P_s} = 0.1765\left[\frac{1}{(1-0.1)^3+0.176} + \frac{2(0.707)}{(1-0.0707)^3+0.176} - \frac{2(0.707)}{(1.0707)^3+0.176} - \frac{1}{(1.1)^3+0.176}\right]$$

$$\frac{R}{a_f A_p P_s} = 0.155$$

FOR $\epsilon = 0.2$

$$\frac{R}{a_f A_p P_s} = 0.1765\left[\frac{1}{(1-0.2)^3+0.176} + \frac{2(0.707)}{(1-.141)^3+0.176} - \frac{2(0.707)}{(1.141)^3+0.176} - \frac{1}{(1.2)^3+0.176}\right]$$

$$\frac{R}{a_f A_p P_s} = 0.331$$

FOR $\epsilon = 0.3$

$$\frac{R}{a_f A_p P_s} = 0.1765\left[\frac{1}{(1-0.3)^3+0.176} + \frac{2(0.707)}{(1-0.212)^3+0.176} - \frac{2(0.707)}{(1.212)^3+0.176} - \frac{1}{(1.3)^3+0.176}\right]$$

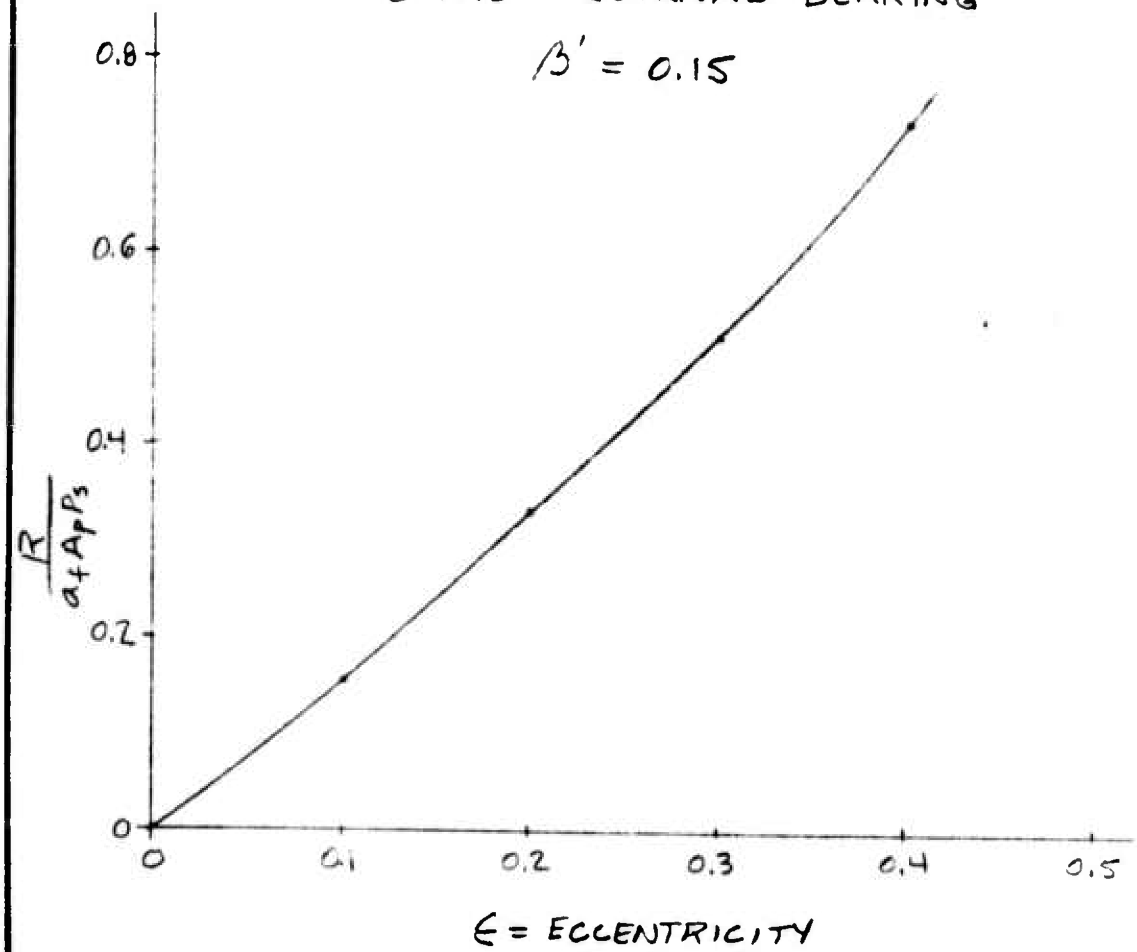
$$\frac{R}{a_f A_p P_s} = 0.513$$

FOR $\epsilon = 0.4$

$$\frac{R}{a_+ A_p P_s} = \left[\frac{1}{(1-.4)^3 + 0.176} + \frac{2(0.707)}{(1-.283)^3 + 0.176} - \frac{2(0.707)}{(1.283)^3 + 0.176} - \frac{1}{(1.4)^3 + 0.176} \right]$$

$$\frac{R}{a_+ A_p P_s} = 0.718$$

LOAD DISPACEMENT CURVE FOR
8 PAD JOURNAL BEARING
$\beta' = 0.15$



FOR GREATER STABILITY USE AN ECCENTRICITY OF APPROXIMATELY

$$\epsilon = 0.3$$

WITH A VERTICAL BEARING LOAD

$$R = 5000 \text{ LB}$$

AND RADIAL CLEARANCE

$$C = 0.003$$

$$e = \epsilon C = 0.3 \times 0.003$$

$$e = 0.0009 \text{ IN}$$

APPROXIMATE BEARING STIFFNESS

$$S = \frac{R}{e} = \frac{5000}{0.0009}$$

$$S \approx 5.5 \times 10^6 \text{ LB/IN}$$

SIZE PADS FOR ABOVE CHARACTERISTICS

$$\text{for } \epsilon = 0.3 \quad \frac{R}{a_f A_p P_s} = 0.513$$

$$a_f A_p = \frac{R}{0.513 (P_s)}$$

NORMAL OPERATING PRESSURE $P_s = 230$ PSI

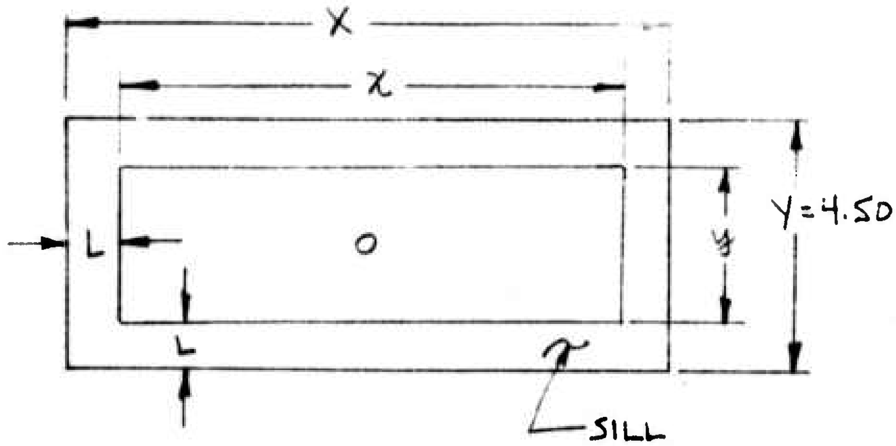
OWENS-ILLINOIS
FECKER SYSTEMS DIVISION

ENGINEERING DATA

$$a_f A_p = \frac{5000}{0.513 \times 230}$$

$$a_f A_p = 42.4 \text{ IN}^2$$

AVAILABLE PAD WIDTH $Y = 4.50$ IN



LET $\frac{X}{Y} = 3.00$

THEN $X = 13.50$ IN

PAD AREA $A_p = XY = 13.50 \times 4.50$

$$A_p = 60.8 \text{ IN}$$

$$a_f = \frac{42.4}{A_p} = \frac{42.4}{60.8}$$

$$a_f = 0.698$$

SUBJECT
BY DATE
PAGE

FROM REF DESIGN MANUAL BY H. RIPPEL

$$\frac{y}{Y} = 0.53$$

FOR $Y = 4.50$ IN

$y = 2.38$ IN

BY DEFINITION OF PAD GEOMETRY

$$Y - y = X - x = 2L$$

$$x = X - (Y - y) = 13.50 - (4.50 - 2.38)$$

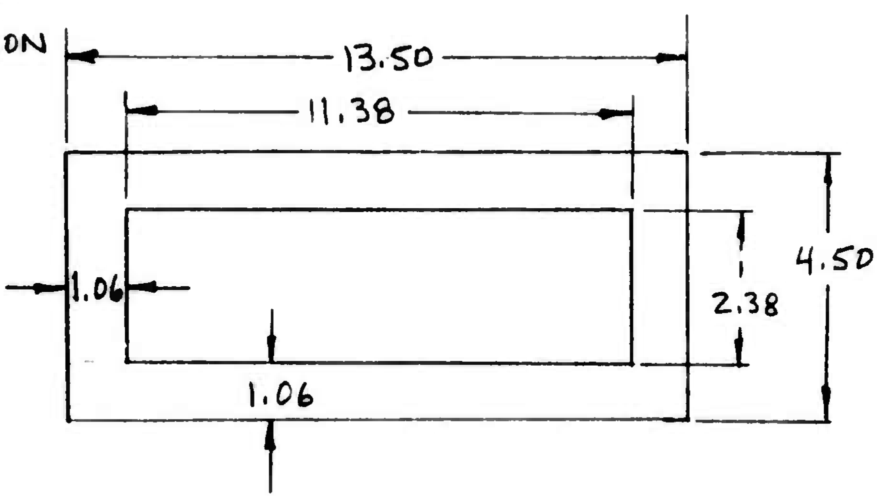
$$x = 11.38$$

SILL WIDTH L

$$L = \frac{Y-y}{2} = \frac{X-x}{2} = \frac{2.12}{2}$$

$$L = 1.06 \text{ IN}$$

FINAL PAD CONFIGURATION



PAD PRELOAD

$$W' = a_f A_p P_r' = a_f A_p \beta' P_s$$

$$a_f A_p = 42.4 \text{ IN}^2$$

$$\beta' = 0.15$$

$$P_s = 230 \text{ PSI}$$

$$W' = (42.4)(0.15)(230)$$

$$W' = 1463 \text{ LB}$$

PAD PRELOAD FLOW (i.e. @ R=0)

$$\text{for } \frac{X}{Y} = 3.0 \qquad \text{AND } \frac{y}{Y} = 0.53$$

$$q_f = 3.40$$

$$Q' = q_f \frac{W'}{A_p} \frac{h'^3}{\mu}$$

MINIMUM EXPECTED VISCOSITY $\mu = 4 \times 10^{-6}$ REYN

$$Q' = 3.40 \left(\frac{1463}{60.8}\right)\left(\frac{0.003^3}{4 \times 10^{-6}}\right)$$

$$Q' = 0.552 \text{ IN}^3/\text{SEC}$$

INDIVIDUAL PAD PARAMETERS

FILM THICKNESS

$$h_n = C - e \cos\theta_n$$

$$C = 0.003$$

$$e = 0.0009$$

$h_1 = 0.003 - 0.0009(1) = 0.$[illegible]$21$ IN

$h_2 = 0.003 - 0.0009(0.707) = 0.00236$ IN

$h_3 = 0.003 - 0.0009(0) = 0.003$ IN

$h_4 = 0.003 - 0.0009(-0.707) = 0.00364$ IN

$h_5 = 0.003 - (0.009)(-1) = 0.0039$ IN

$h_6 = 0.003 - (0.0009)(-0.707) = 0.00364$ IN

$h_7 = 0.003 - (0.0009)(0) = 0.003$ IN

$h_8 = 0.003 - (0.0009)(0.707) = 0.00236$ IN

INDIVIDUAL PAD PARAMETERS

PAD FLOWS

$$\frac{Q_n}{Q'} = \frac{\frac{1}{\beta'}(1-\epsilon\cos\theta)^3}{\left(\frac{1}{\beta'}-1\right)(1-\epsilon\cos\theta)^3+1} = \frac{6.67(1-\epsilon\cos\theta)^3}{5.67(1-\epsilon\cos\theta)^3+1}$$

$$\epsilon = 0.3$$

$$Q' = 0.552\ IN^3/SEC$$ (AT MINIMUM VISCOSITY — I.E. MAXIMUM FLOW)

PAD #1

$$\frac{Q_1}{Q'} = \frac{6.67(1-0.3)^3}{5.67(1-0.3)^3+1} = \frac{6.67(0.343)}{5.67(0.343)+1} = \frac{2.29}{2.95} = 0.775$$

$$Q_1 = 0.428\ IN^3/SEC$$

PAD #2

$$\frac{Q_2}{Q'} = \frac{6.67(1-0.212)^3}{5.67(1-0.212)^3+1} = \frac{6.67(0.489)}{5.67(0.489)+1} = \frac{3.26}{3.77} = 0.865$$

$$Q_2 = 0.477\ IN^3/SEC$$

PAD #3

$$\frac{Q_3}{Q'} = \frac{6.67(1-0)^3}{5.67(1-0)^3+1} = \frac{6.67}{6.67} = 1.0$$

$$Q_3 = 0.552\ IN^3/SEC$$

PAD #4

$$\frac{Q_4}{Q'} = \frac{6.67(1+0.212)^3}{5.67(1+0.212)^3+1} = \frac{6.67(1.78)}{5.67(1.78)+1} = \frac{11.88}{11.1} = 1.07$$

$$Q_4 = 0.591\ IN^3/SEC$$

INDIVIDUAL PAD PARAMETERS

PAD FLOWS

PAD #5

$$\frac{Q_5}{Q'} = \frac{6.67(1+0.3)^3}{5.57(1+0.3)^3+1} = \frac{6.67(2.195)}{5.67(2.195)+1} = \frac{14.62}{13.32} = 1.10$$

$$Q_5 = 0.606 \text{ IN}^3/\text{SEC}$$

PAD #6

$$Q_6 = Q_4 = 0.591$$

PAD #7

$$Q_7 = Q_3 = 0.552$$

PAD #8

$$Q_8 = Q_2 = 0.477 \text{ IN}^3/\text{SEC.}$$

TOTAL FLOW - ALL PADS

$$Q_T = Q_1 + Q_2 + Q_3 + \cdots Q_8$$
$$= 0.428 + 2 \times 0.477 + 2 \times 0.552 + 2 \times 0.591 + 0.606$$
$$Q_T = 3.668 \text{ IN}^3/\text{SEC } (0.955 \text{ GPM})$$

FLOW HORSEPOWER

$$H_P = \frac{3.668 \times 230}{6600} = 0.128 \text{ hp}$$

INDIVIDUAL PAD PARAMETERS

PAD RECESS PRESSURES

$$P_{r_1} = \frac{\mu \; Q_1}{q_f \, a_f \, h^3} = \frac{4 \times 10^{-6}}{(3.4)(0.698)} \frac{Q_1}{h_1^3} = 1.69 \times 10^{-6} \frac{.428}{0.0021^3} = 78.2 \text{ PSI}$$

$$P_{r_2} = 1.69 \times 10^{-6} \times \frac{Q_2}{h_2^3} = 1.69 \times 10^{-6} \frac{0.477}{0.00236^3} = 61.5 \text{ PSI}$$

$$P_{r_3} = 1.69 \times 10^{-6} \frac{0.552}{0.003^3} = 34.5 \text{ PSI}$$

$$P_{r_4} = 1.69 \times 10^{-6} \frac{0.591}{0.00364^3} = 20.8 \text{ PSI}$$

$$P_{r_5} = 1.69 \times 10^{-6} \frac{0.606}{0.0039^3} = 16.5 \text{ PSI}$$

$$P_{r_6} = P_{r_4} = 20.8 \text{ PSI}$$

$$P_{r_7} = P_{r_3} = 34.5 \text{ PSI}$$

$$P_{r_8} = P_{r_2} = 61.5 \text{ PSI}$$

PRESSURE RATIO

$$\beta_1 = \frac{P_{r_1}}{P_s} = 0.34 \qquad \beta_5 = 0.072$$

$$\beta_2 = 0.267 \qquad \beta_6 = 0.091$$

$$\beta_3 = 0.150 \qquad \beta_7 = 0.150$$

$$\beta_4 = 0.091 \qquad \beta_8 = 0.267$$

INDIVIDUAL PAD PARAMETERS

PAD LOADS

$W_1 = a_f A_p Pr_1 = (0.698)(60.8)(78.2) = 3320$ LB

$W_2 = 42.4 \times 61.5 = 2510$ LB

$W_3 = 42.4 \times 34.5 = 1463$ LB

$W_4 = 42.4 \times 20.8 = 882$ LB

$W_5 = 42.4 \times 16.5 = 700$ LB

$W_6 = 882$ LB

$W_7 = 1463$ LB

$W_8 = 2510$ LB

SIZING OF CAPILLARY RESTRICTORS

EQUATE PAD FLOW TO CAPILLARY FLOW

$$Q' = q_f a_f P_r' \frac{h'^3}{\mu} = \frac{\pi d_c^4 (P_s - P_r')}{128 \mu l_c}$$

WHERE d_c = CAPILLARY TUBE DIAMETER

l_c = CAPILLARY TUBE LENGTH

$$\beta' = \frac{P_r'}{P_s} = 0.15$$

$$\frac{d_c^4}{l_c} = \left(\frac{\beta'}{1-\beta'}\right) \frac{128}{\pi} a_f q_f h'^3$$

$$\frac{d_c^4}{l_c} = \left(\frac{0.15}{1-0.15}\right) \frac{128}{\pi} (0.698)(3.110)(0.003)^3$$

$$\frac{d_c^4}{l_c} = 46.0 \times 10^{-8}$$

USING STANDARD TUBING DIAMETER $d_c = 0.035$ IN

$$l_c = \frac{(0.035)^4}{46.0 \times 10^{-8}} = \frac{184}{46}$$

$$l_c = 4.00$$

OWENS-ILLINOIS
FECKER SYSTEMS DIVISION

ENGINEERING DATA

CHECK MAXIMUM REYNOLDS NUMBER

MAXIMUM FLOW OCCURS AT PAD #5

$$Q_5 = 0.606 \text{ IN}^3/\text{SEC}$$

$$N_R = \frac{4 \rho Q_5}{\pi d_c \mu}$$

$$N_R = \frac{4 \times 89.3 \cancel{\times 10^{-6}} \times 0.606}{\pi (0.035)(4 \times 10^{-6})}$$

$$N_R = 492 \quad < 2000 \quad \text{OK}$$

SUBJECT
BY DATE
PAGE